ENGINEERING DIGEST:
MR. T.B. WEBB
MR. W.R. HARVEY
MR. F.S. TOLLOW

The ENGINEERING DIGEST

REVIEW OF WORLD ENGINEERING PROGRESS

VOL. XIX No. 7 JULY, 1958

Complete Hoist Control without Mechanical Braking

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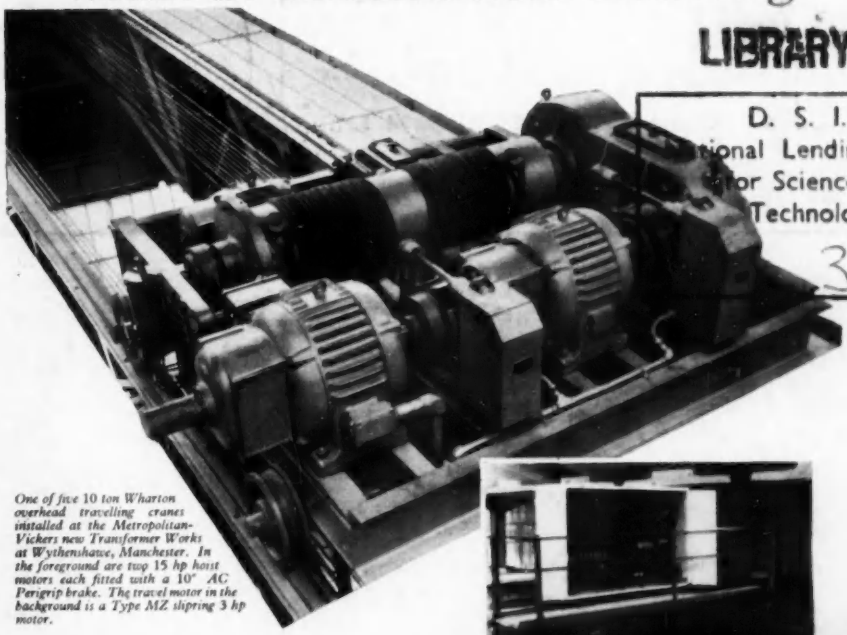
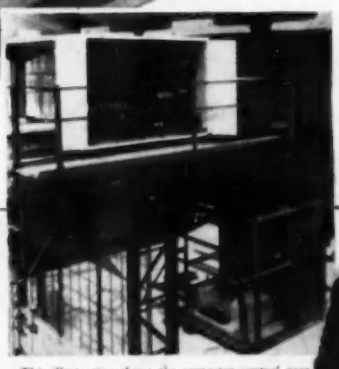


One of five 10 ton Wharton overhead travelling cranes installed at the Metropolitan-Vickers new Transformer Works at Wythenshawe, Manchester. In the foreground are two 15 hp hoist motors each fitted with a 10" AC Perigrip brake. The travel motor in the background is a Type MZ slipring 3 hp motor.



This illustration shows the contactor control gear the operator's cabin.

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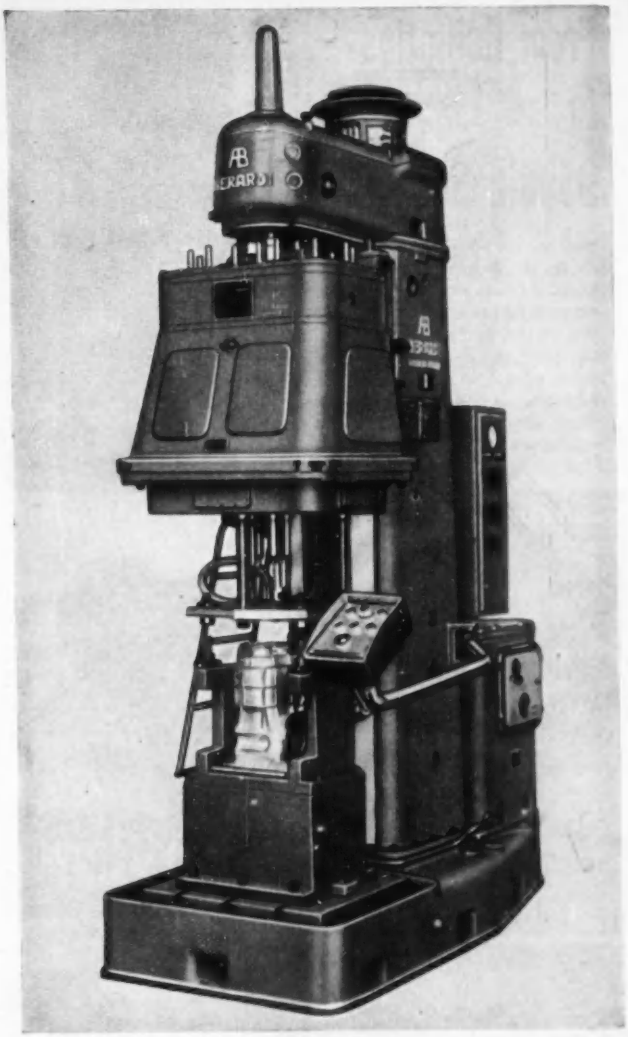

Drilling operations on gear change and differential box for car.

Component referred and clamped in stationary fixture.

Drilling jig fixed to the head.

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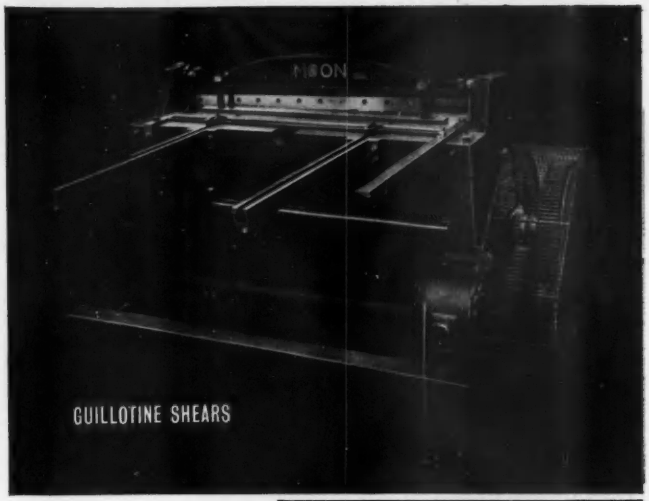

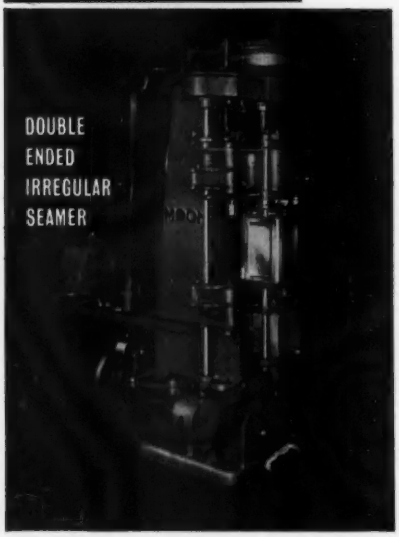

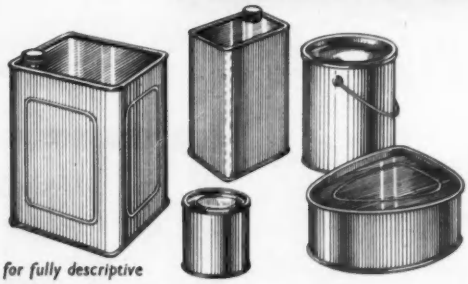

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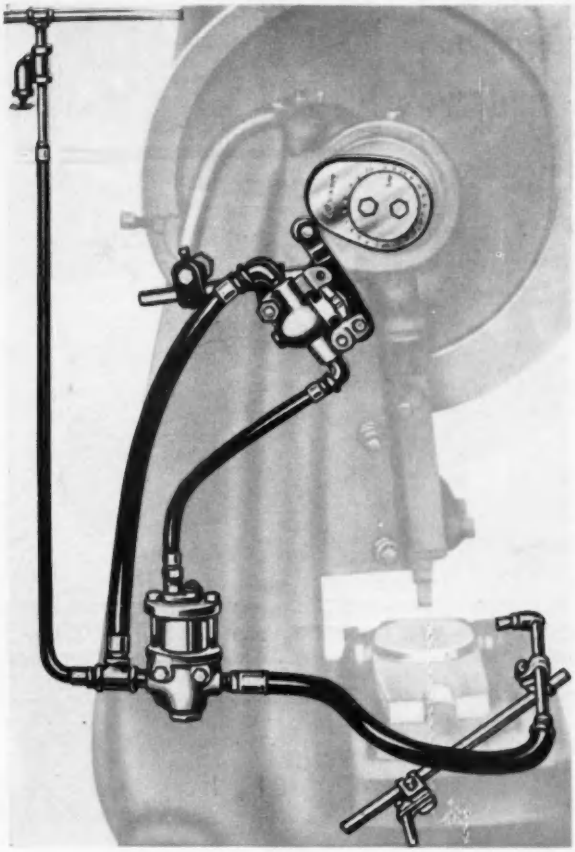

This Air Ejection Set installation shows cam operation off the press crankshaft. It will provide an air blast timed for a fraction of a second, or longer, at any desired point in the press cycle. If required the air blast can be continued after press motion has ceased.

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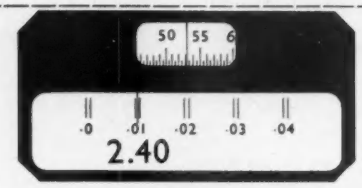

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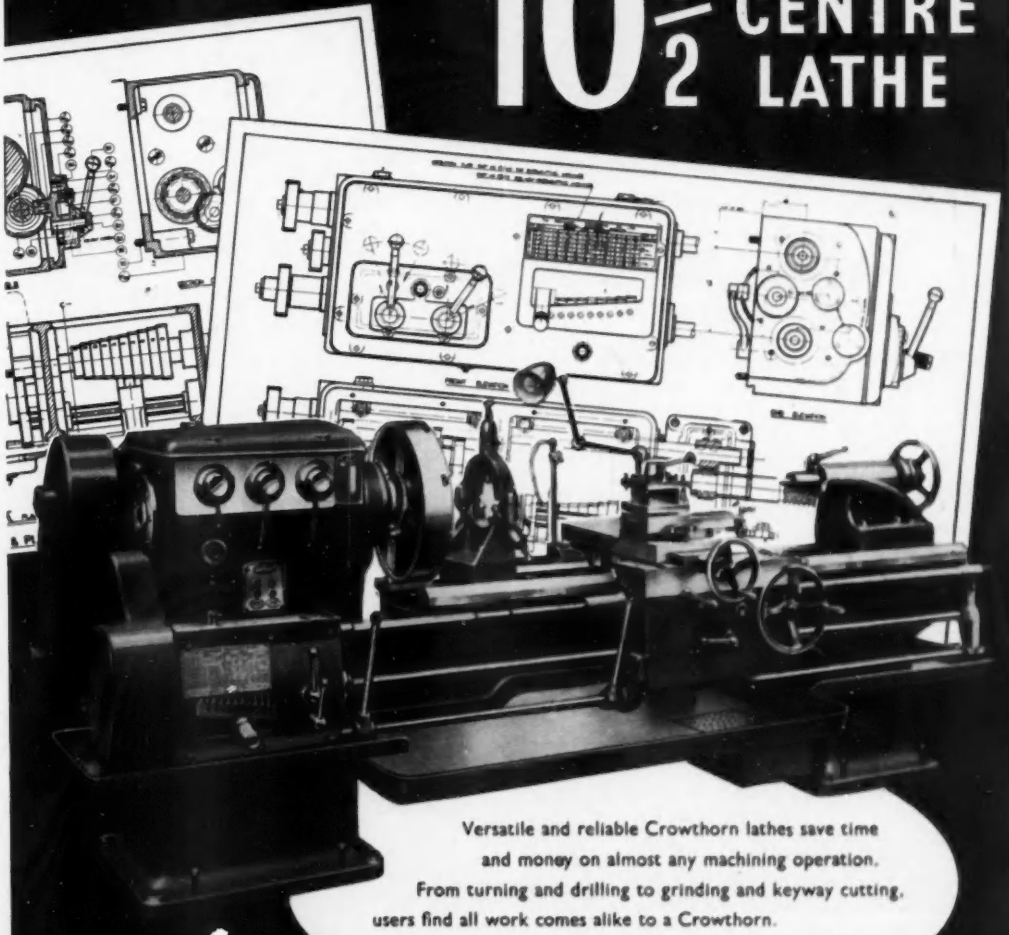

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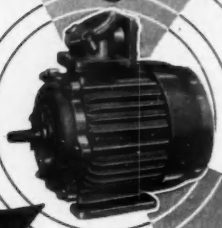

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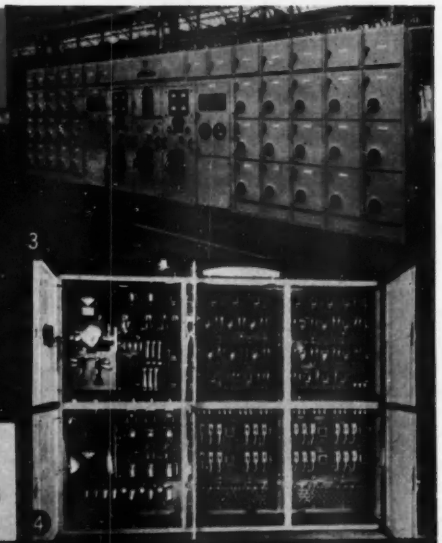

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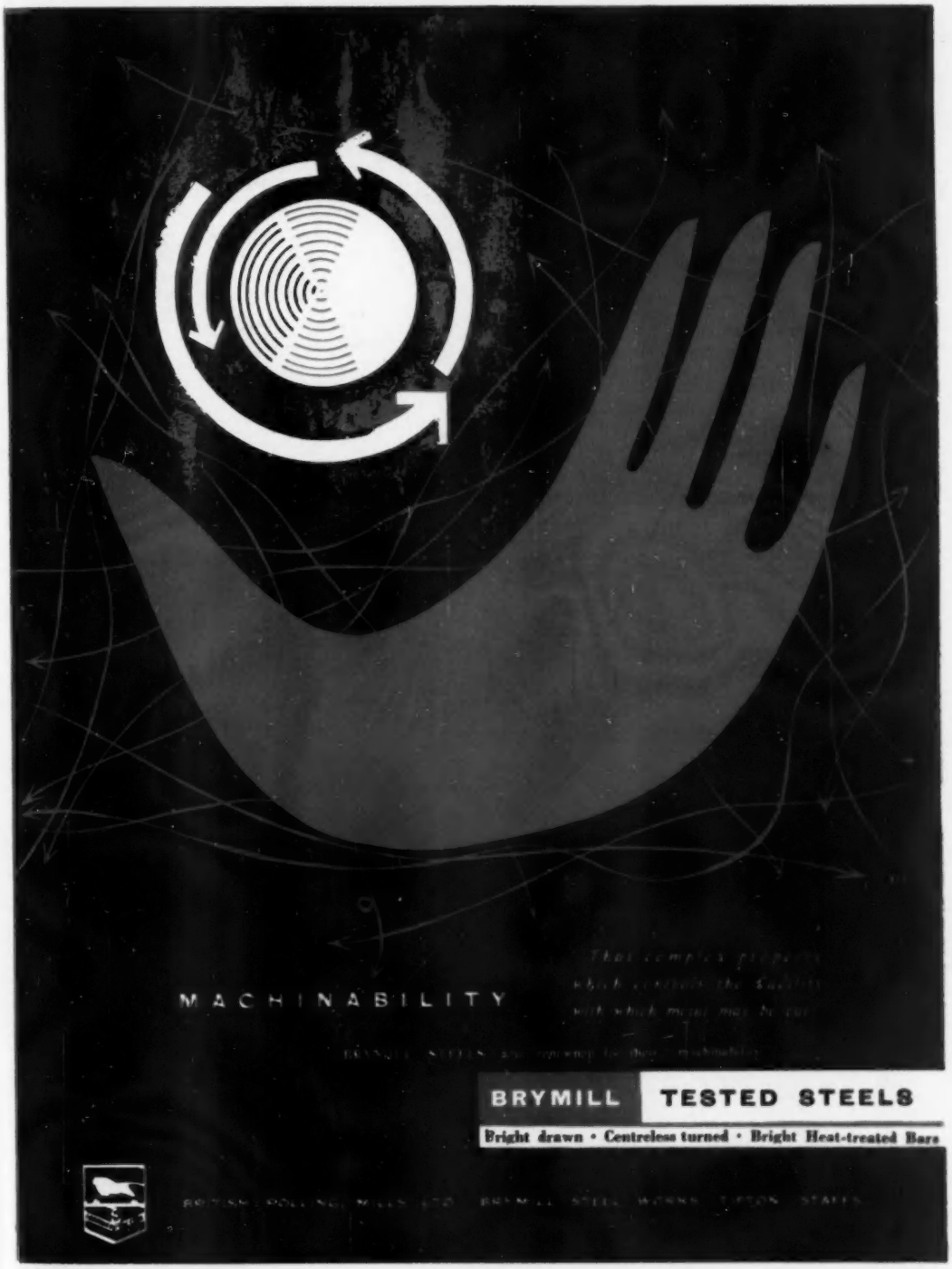
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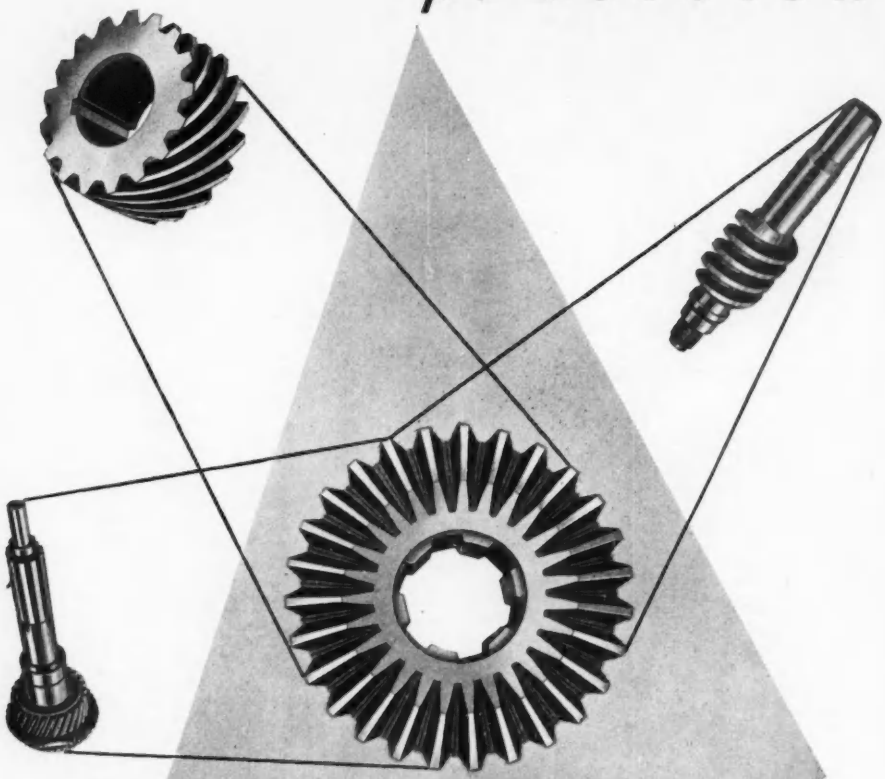

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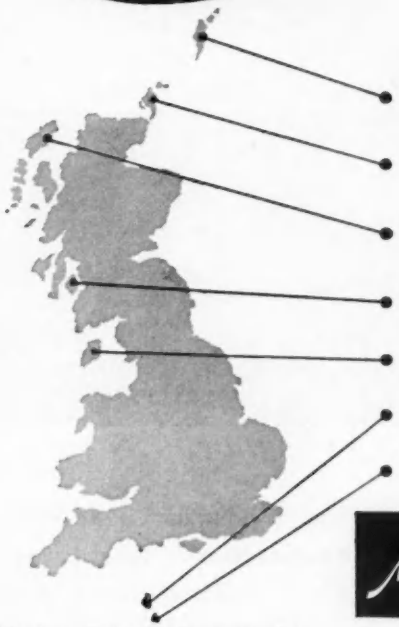

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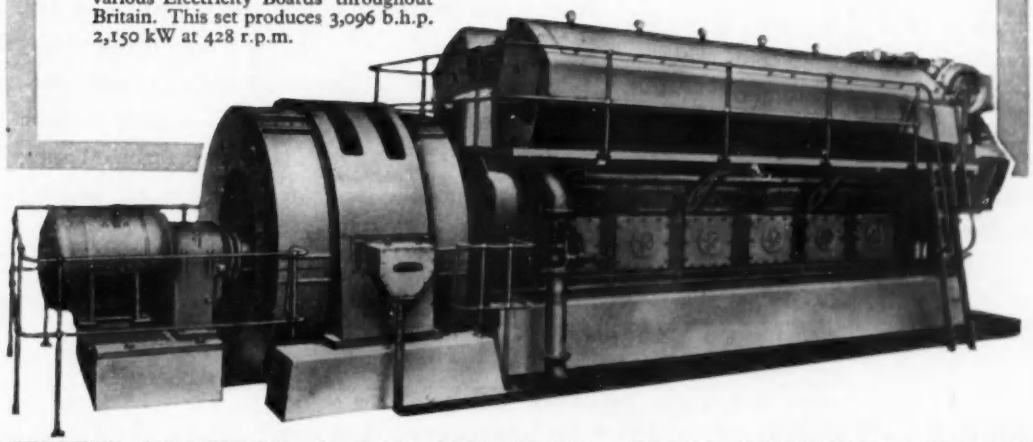

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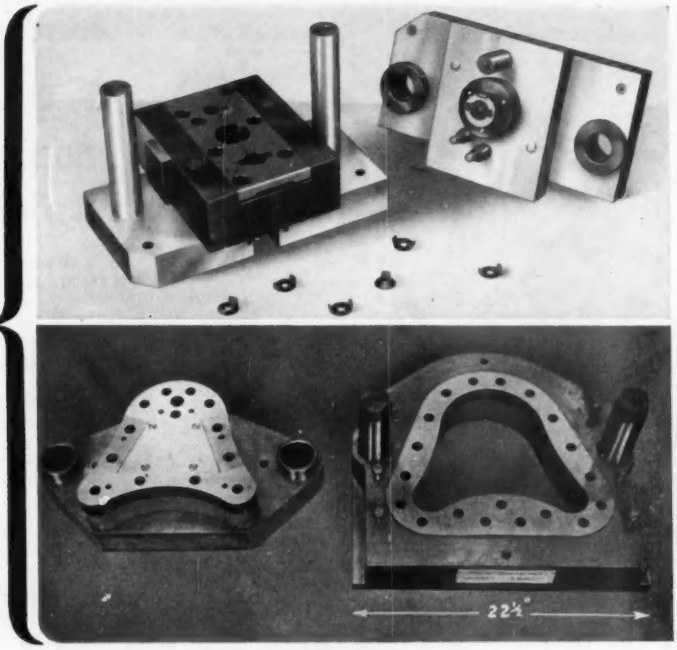

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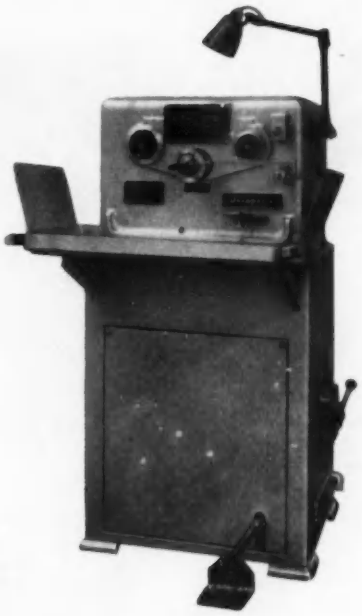

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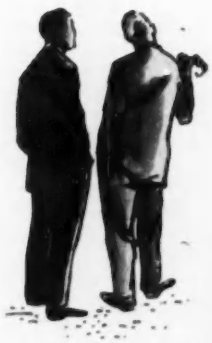

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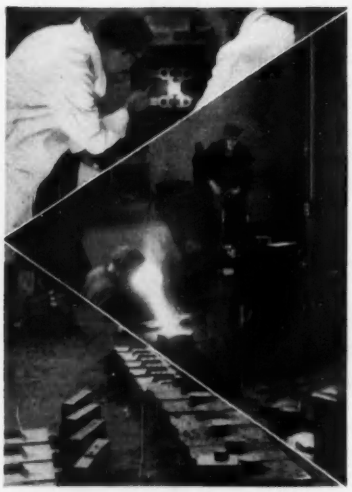

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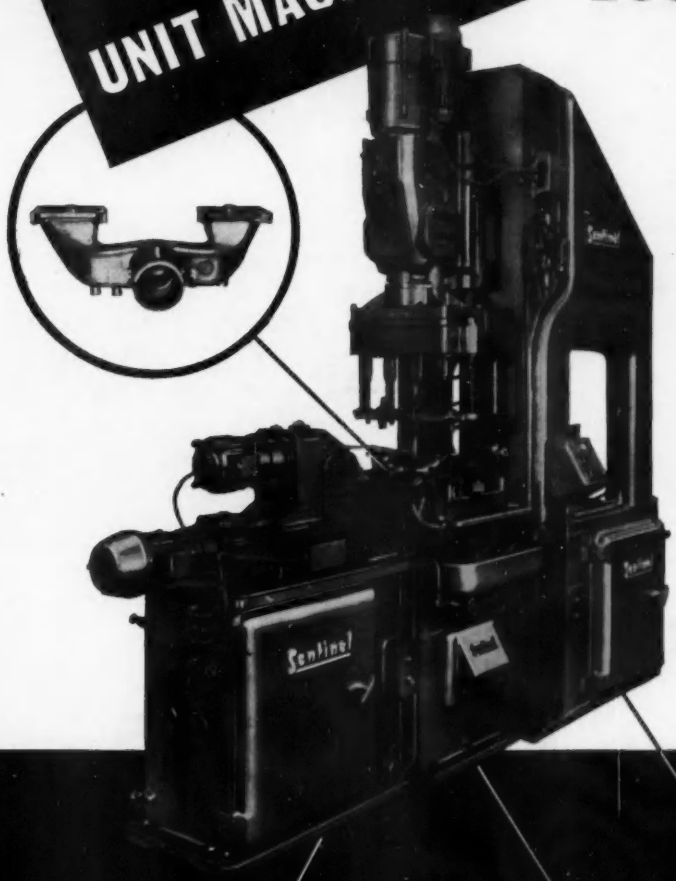

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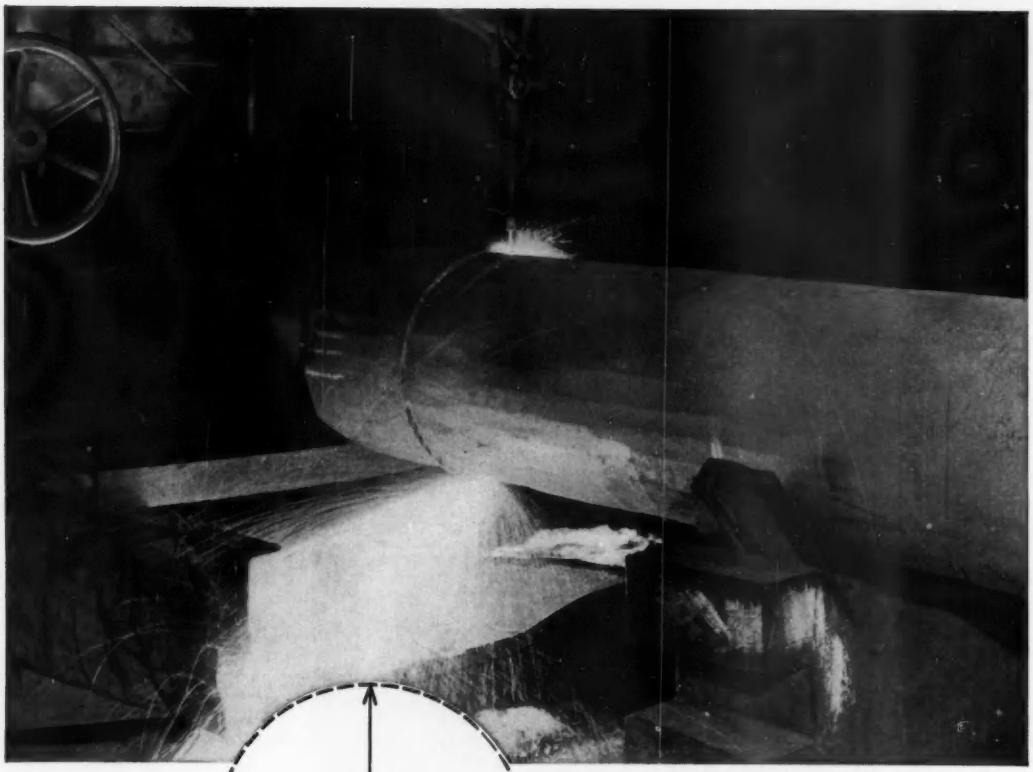

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Holroyd HO	**Single reduction overdriven with horizontal input and output shafts. Centre distances 3, 4, 5, 6, 7, 8, 9, 10 and 12 inches.**
Holroyd HV	**Single reduction with vertical output shaft and horizontal input. Centre distances 3, 4, 5, 6, 7, 8, 9, 10 and 12 inches.**
Holroyd Verso	**2¼ inch centre distance. Suitable for underdriven, overdriven or side mounting.**
Holroyd F	**Made in three sizes—1¼, 1¾ and 2¼ inch centre distance, and suitable for a variety of mounting positions.**
All the above are available with any of the following ratios:—5, 7½, 10, 15, 20, 25, 30, 40, 50, 60 and 70 to 1.	

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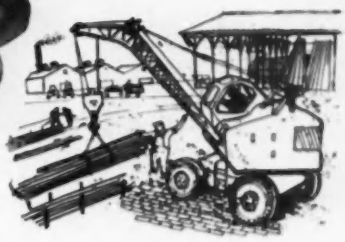

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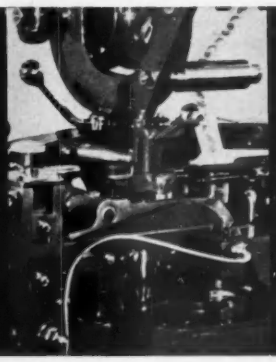

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Flash point—274°C. (525°F.)
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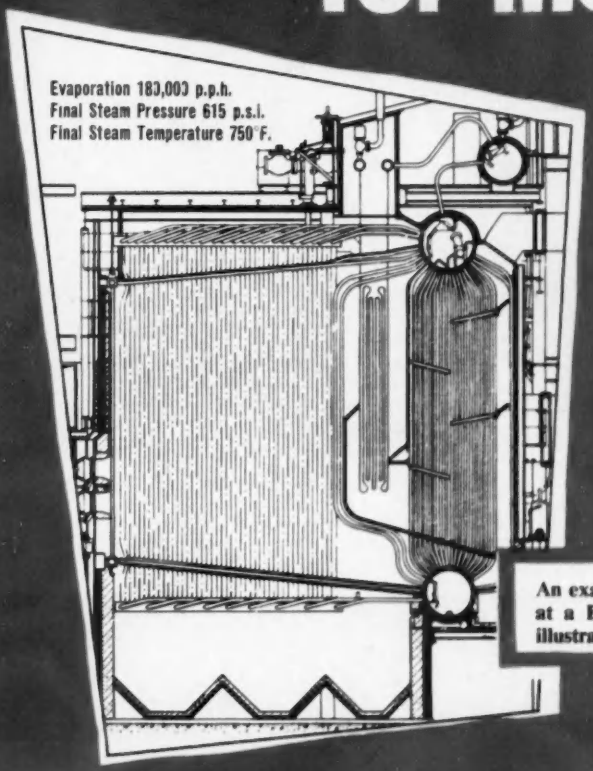

IC build boilers to suit the individual steam raising requirements of every industry.

Whatever the problem, IC steam generating equipment ensures the best use of fuel resources.

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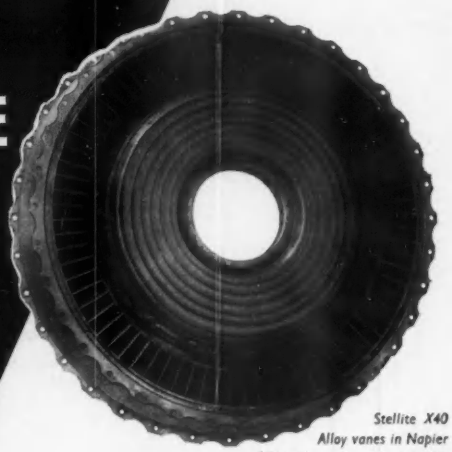

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Alloy vanes in Napier 'Eland' nozzle guide assembly

The resistance to destruction of Deloro Stellite Alloys is worth investigating. Heat, abrasion and corrosion resistance is of a high order. Even combinations of these destructive effects make little impression on these remarkable alloys.

● ***Send for publication SP13 which gives properties of some of the alloys in the wide range.***

The Napier 'Eland' propeller turbine aero engine

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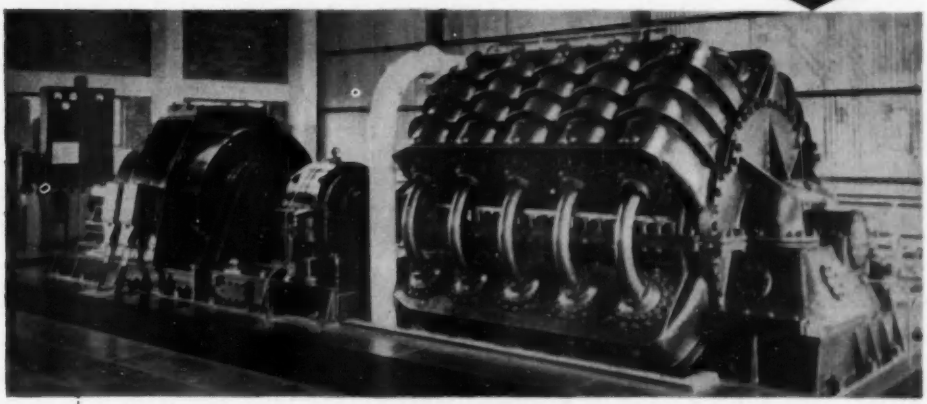
BTH COMPRESSING PLANT AT DRIEFONTEIN GOLD MINE

This BTH motor-driven air compressor supplied to New Consolidated Goldfields Ltd. is now in service at West Driefontein Gold Mine in the Transvaal. The machine* operates at an extremely high efficiency and delivers 20,000 cubic feet a minute at 90 pounds per square inch.

*The design of the machine embodies Oerlikon patents Nos. 54268 and 558910.



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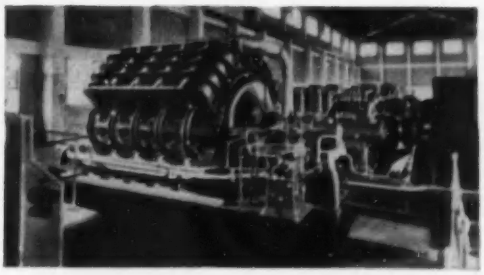

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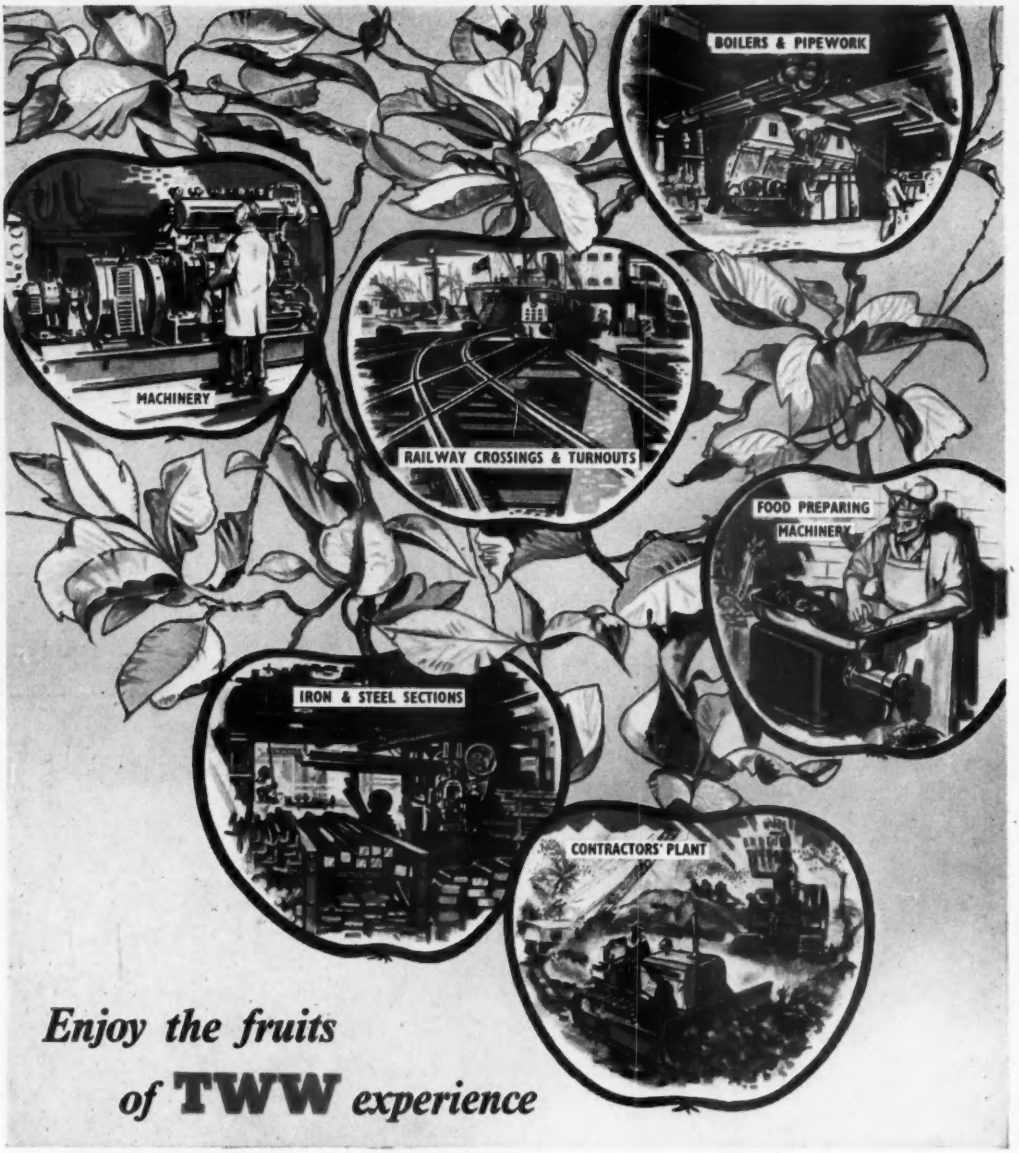

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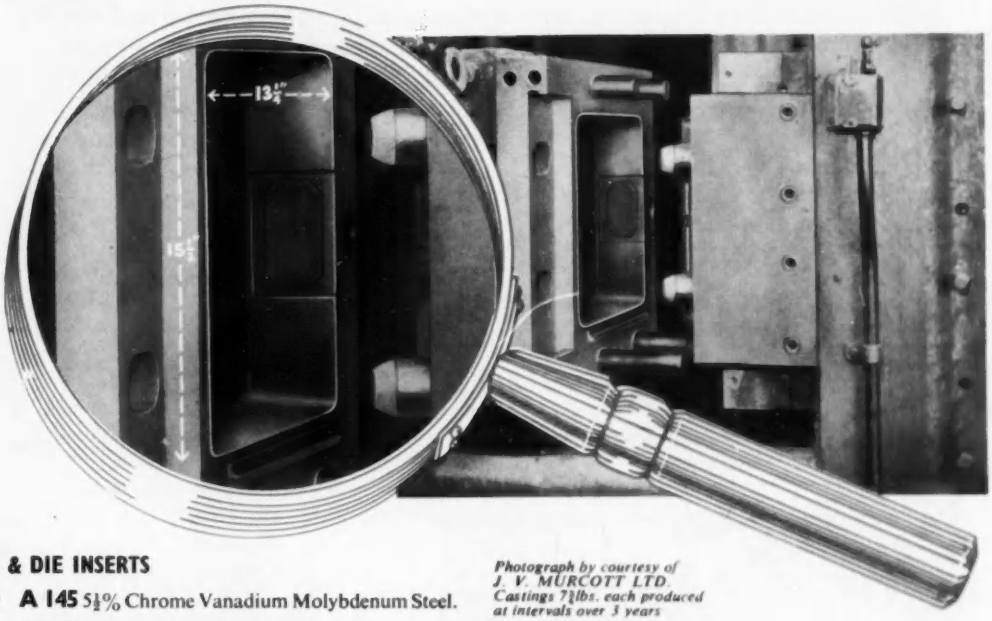

Photograph by courtesy of J. V. MURCOTT LTD. Castings 7¾lbs. each produced at intervals over 3 years

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(KE) **A 145** 5½% Chrome Vanadium Molybdenum Steel.

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(KE) **896** Medium Carbon Chrome Vanadium Steel.

Supplied in ultrasonically tested forged blocks and black bars of square, round and rectangular sections.

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Accuracy, speed, simple operation and robust reliability are typical of the outstanding advantages of this equipment.

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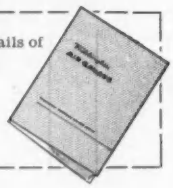

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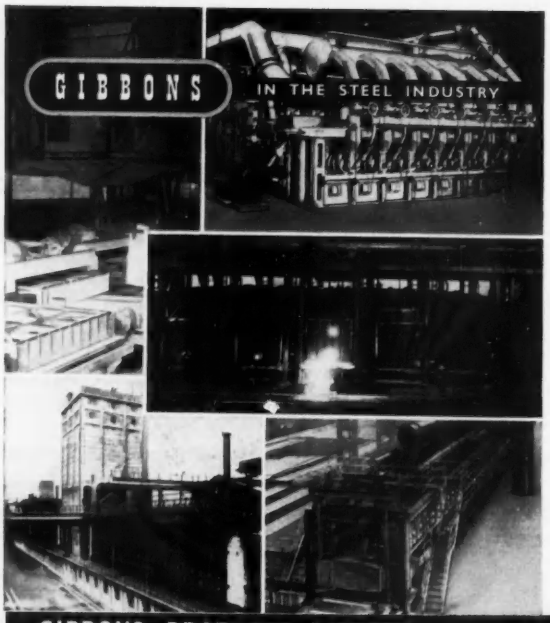

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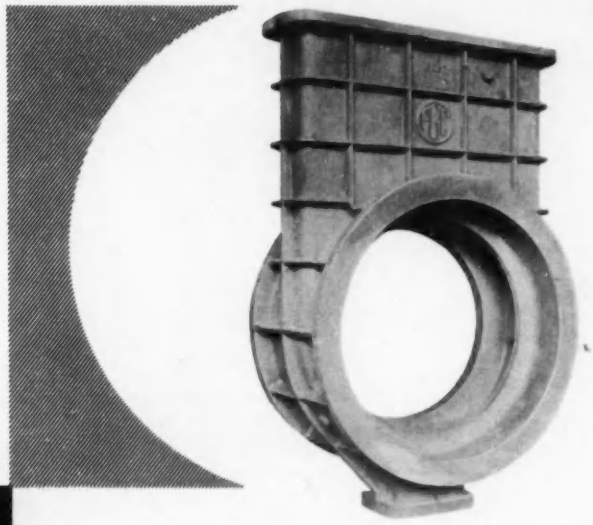

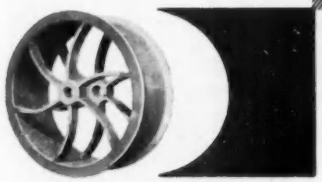

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Ashmore, Benson, Pease & Co. will be pleased to advise the grade suitable for your application.

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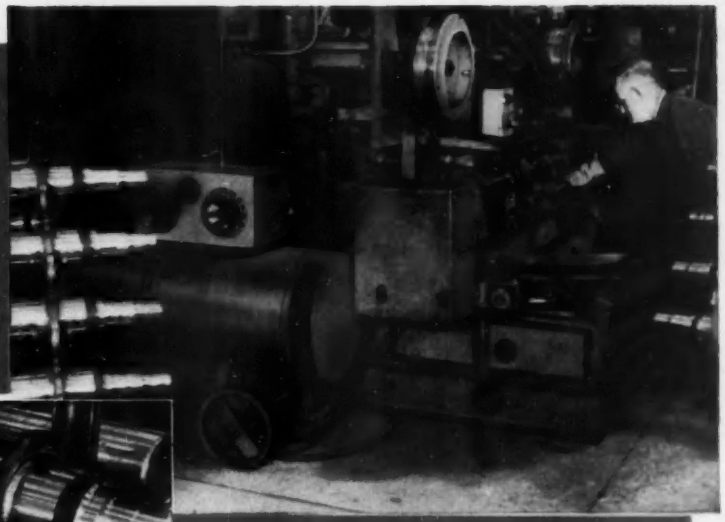

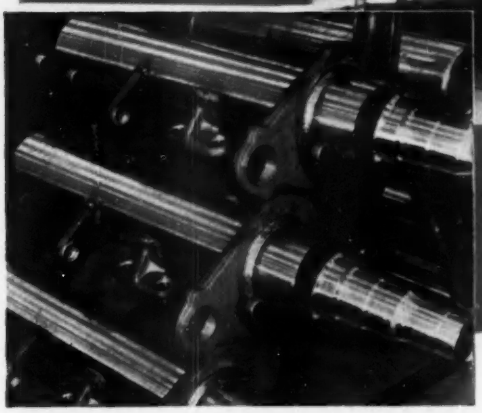

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Table milling feeds ¾″ to 15″ per minute (optional)

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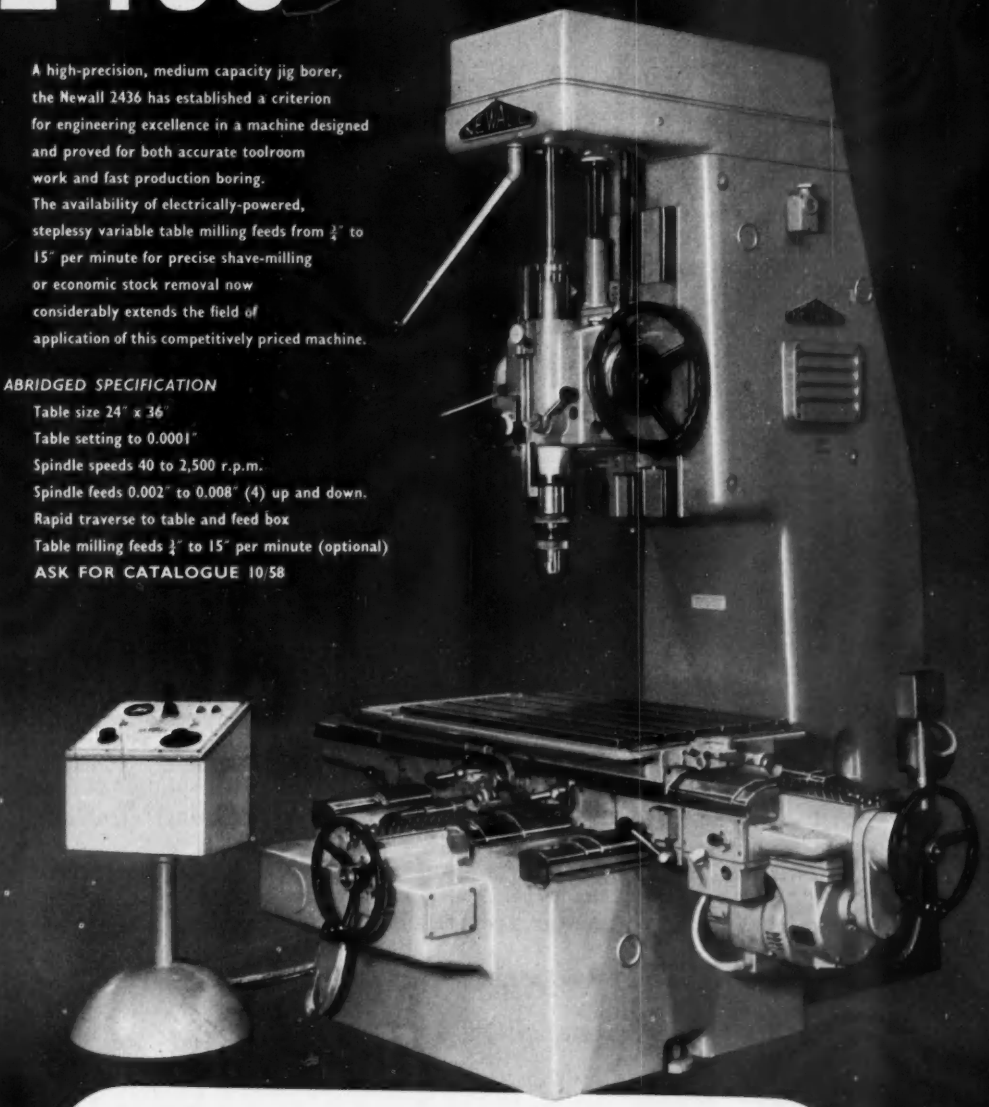

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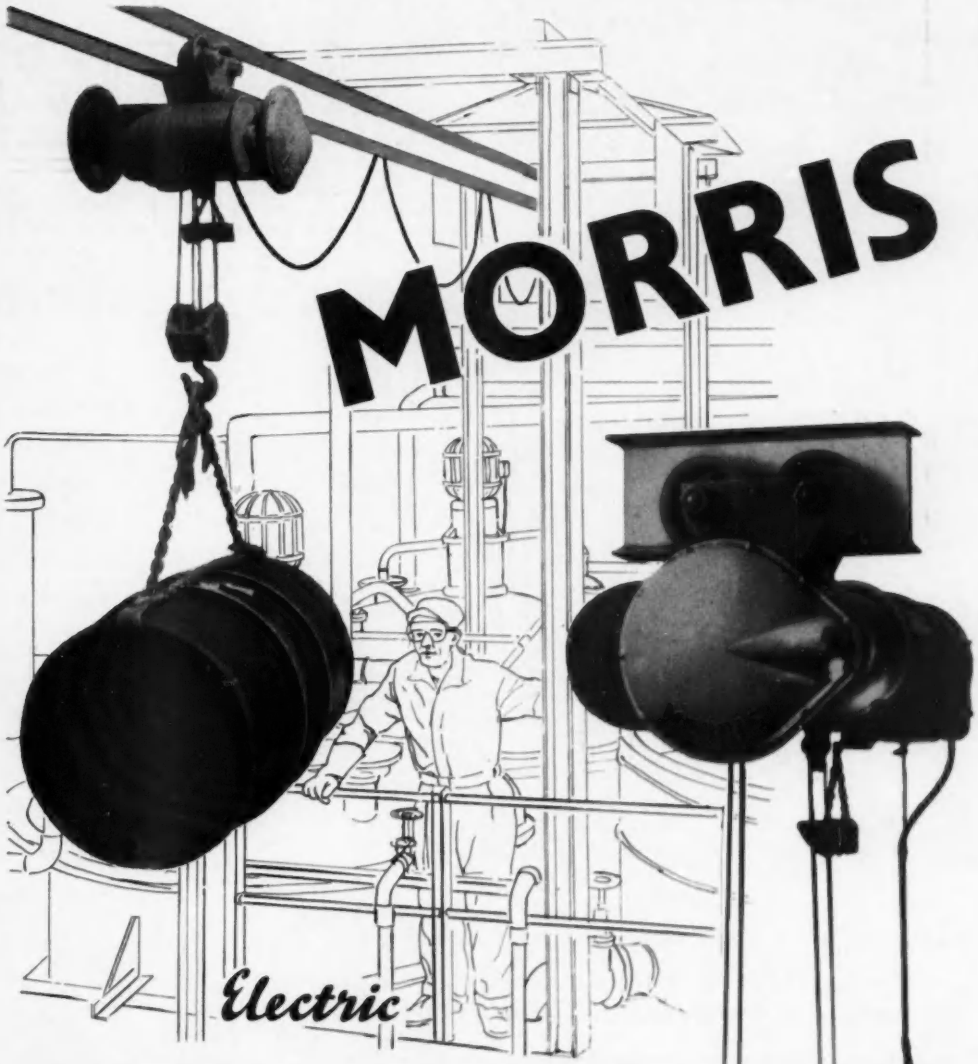

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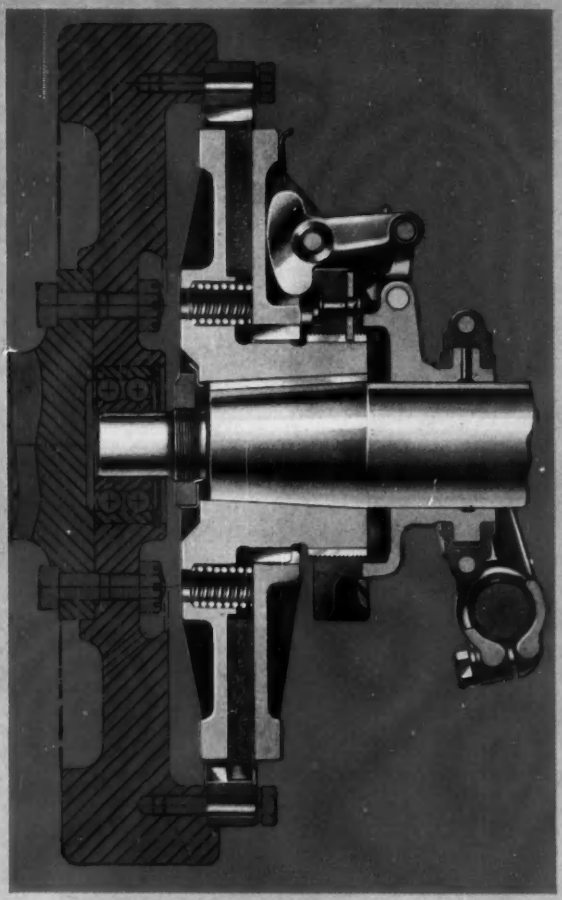

The illustration shows the Perkins 'L.4' diesel engine, with Rockford 'over-centre' clutch, driving through a Cyril Norris reduction gear, giving a low-speed take-off of value in many industries.

BORG & BECK CLUTCHES,
used extensively on cars and commercial vehicles, are made in a wide range of sizes. Particulars of these and Rockford clutches on application.

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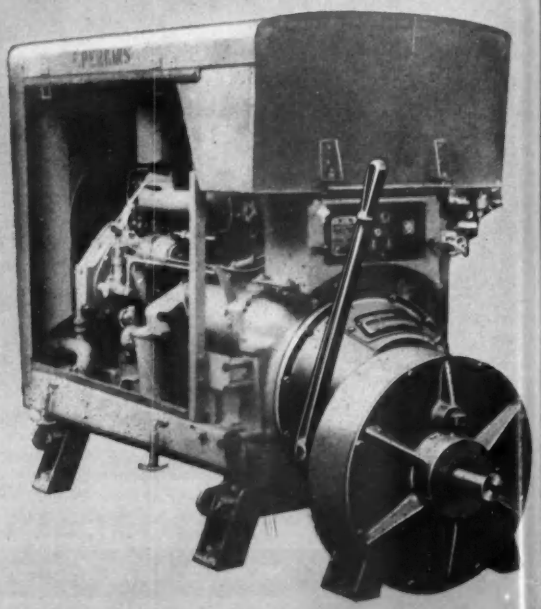

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We do indeed. A bearing's performance is governed by correct lubrication as much as by correct design and materials. The cage location area is one of vital importance. The lubricant must also protect the highly finished surfaces from corrosion and help to dissipate any undue heat. To fulfil these requirements, we have a team of lubrication specialists at work.

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Research into this is continuous. Operating speeds, loads and temperatures are steadily rising and every week brings fresh problems for solution. Aircraft and guided missiles are the obvious—but by no means the only—instances. Speaking generally we recommend lubrication by oil flood or vapour mist for the very highest speeds. We have built up extensive data on this and allied problems. If you get in touch with our Technical Department they'll be only too glad to talk lubrication with you.

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Newark-on-Trent England

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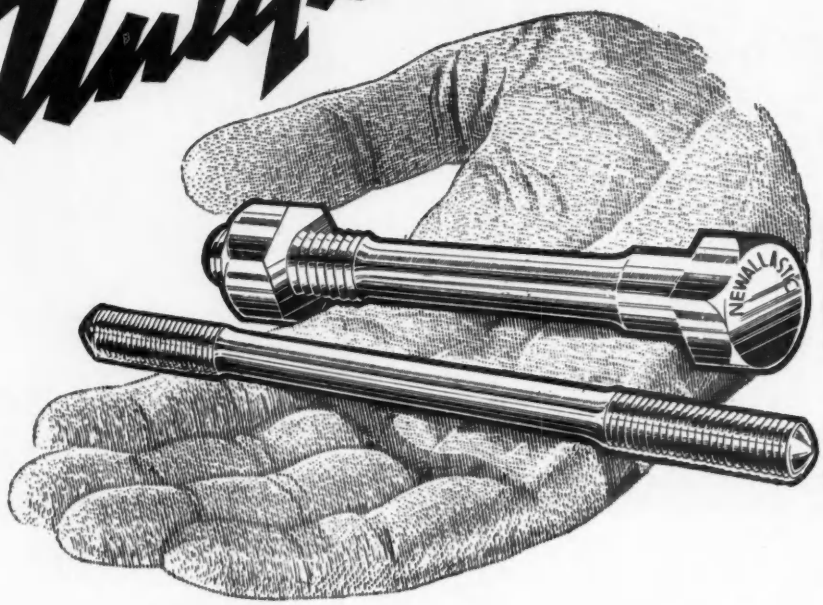

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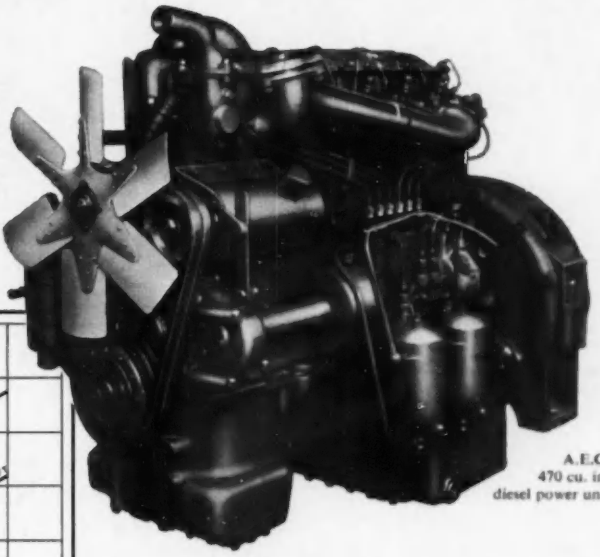
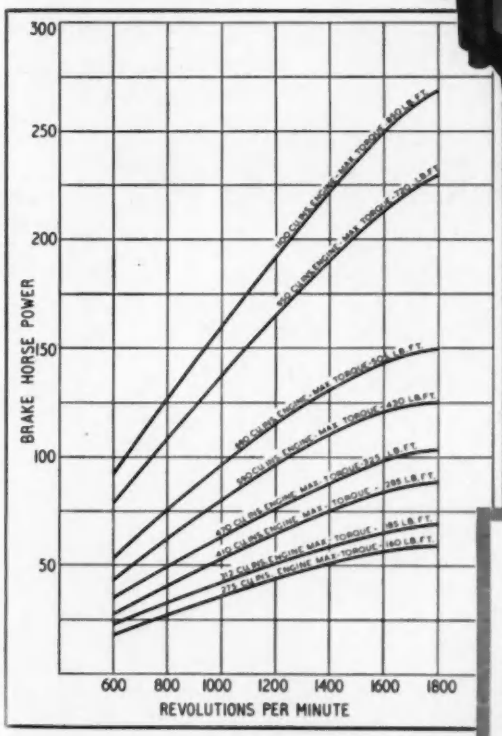


A.E.C. 470 cu. in. diesel power unit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470 CU INS ENGINE - MAX TORQUE 770 LB FT
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- ✱ Ships' Auxiliaries
- ✱ Air Compressors
- ✱ Excavators
- ✱ Tractors and Bulldozers
- ✱ Pumping Sets
- ✱ Diesel Trains
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1/446

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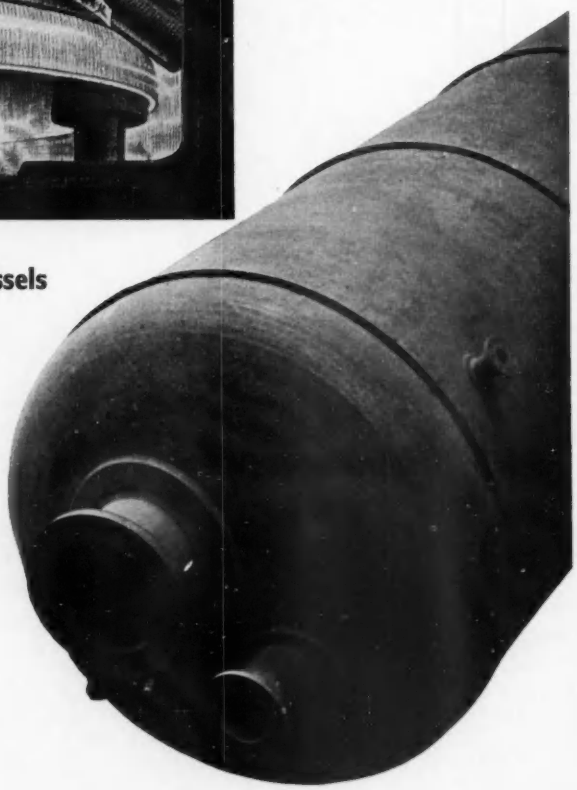

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can be supplied in Mild, Alloy and Clad Steels and Non-ferrous Metals. The capacity of the Rotarpress ranges from 5 to 15 feet diameter and $\frac{3}{8}$" to 4" thickness. Knuckle radii and depth may be varied to meet individual requirements.

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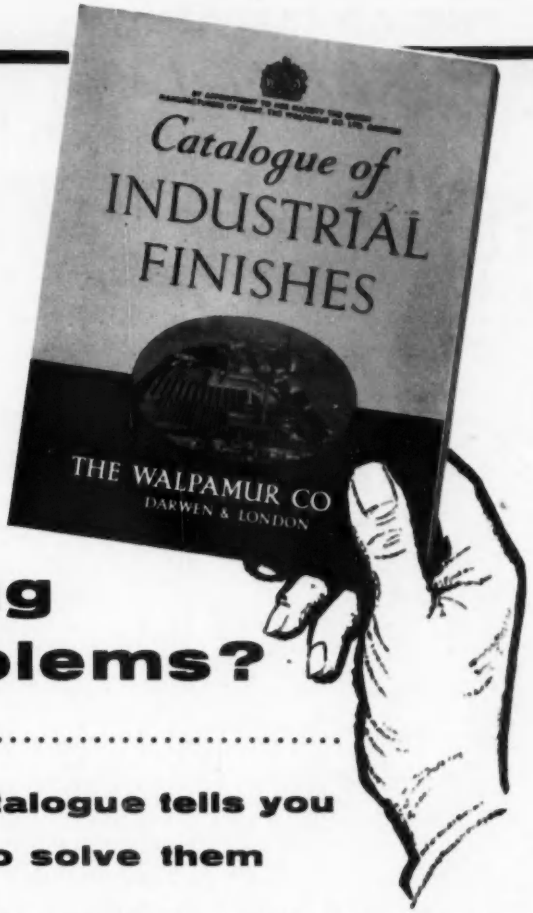

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This FREE catalogue tells you how to solve them

This catalogue contains information about the variety and uses of industrial finishes made by The Walpamur Company Limited, many of which are specially effective in the prevention and control of corrosion of industrial steelwork and plant. Comprehensive data is given regarding the applications and performances of primers for metal, primer fillers and stoppers, low bake and special stoving finishes, quick-drying enamels, and nitro cellulose finishes for wood and metal. Booklets about FEROX Epoxy Resin Paints and Silicone Paints, containing a wealth of information and examples of their uses, and a new Industrial Shade Card showing the full range of colours, are also available on request.

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MANUFACTURERS OF PAINT

THE WALPAMUR COMPANY LIMITED
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Paints, enamels, varnishes and finishes for every industrial need

RAPID THREAD REPAIR

CUTS SCRAP COSTS!

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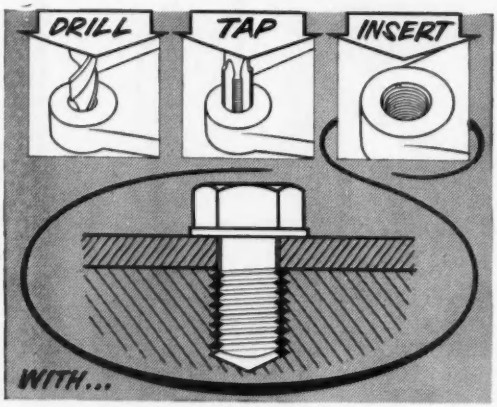

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- Replaces leading—no lead poisoning, non-injurious

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Double Bond has a wide range of production and trade applications: practically every industry can benefit from the saving in time and costs which its use ensures.

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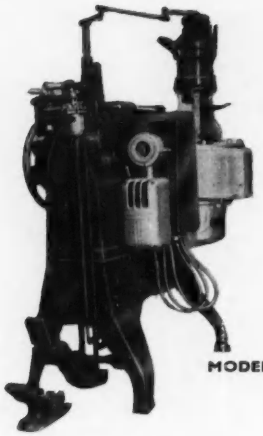

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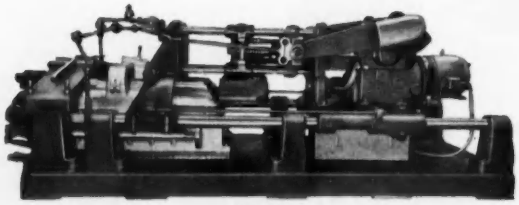

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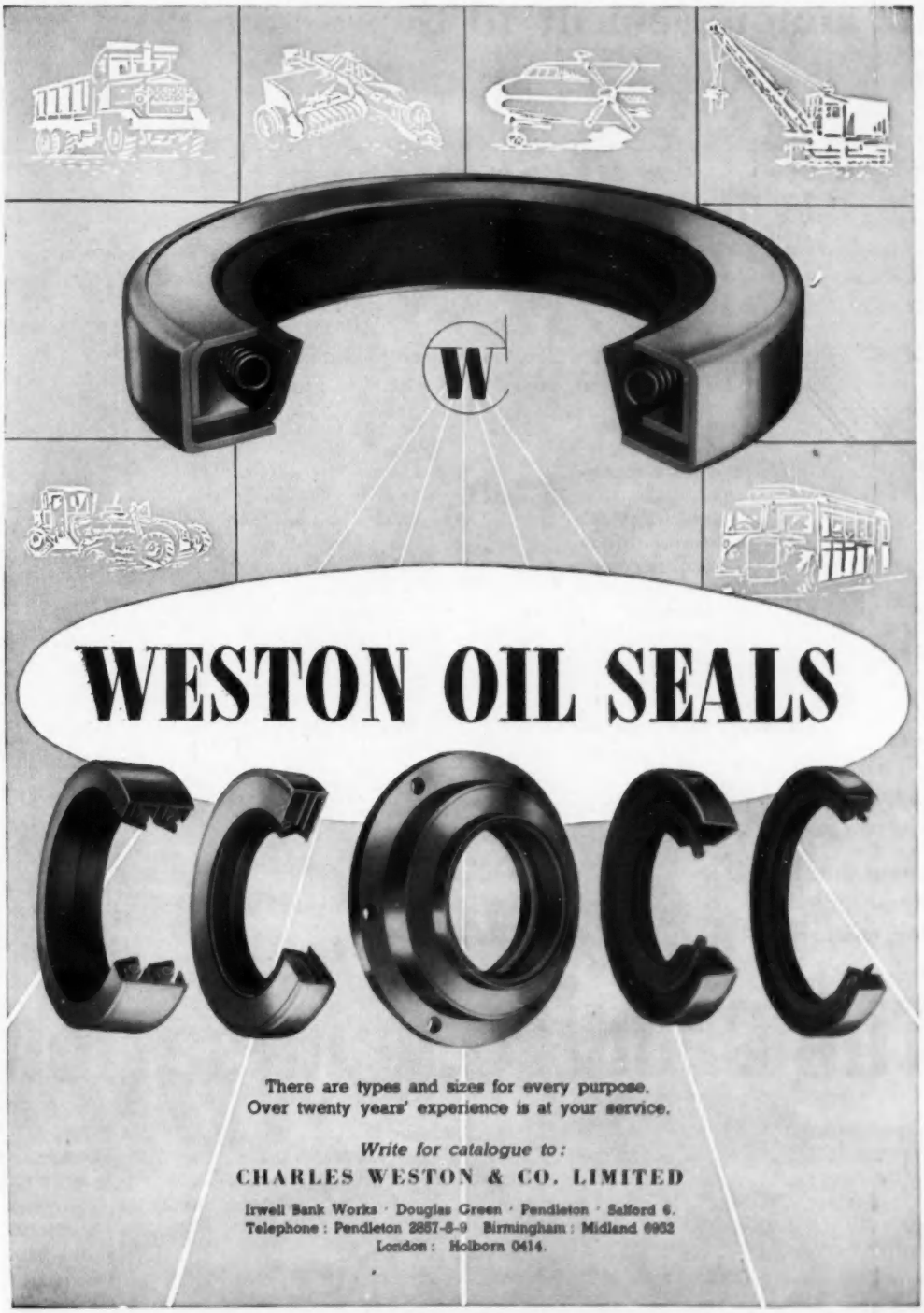
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Anti-corrosive paints
Anti-freeze solutions
Bituminous solutions
Brake fluids
Corrosion inhibitors
Chrome protectives
Cutting oils
Coolant soluble oils
Carton waxes
Drawing oils
De-watering oils
De-icing fluids
De-greasing fluids
Emulsion degreasers
Fingerprint removers
Greases, special
Grease paints
Lanolin solutions
Metal marking inks
Penetrating oils
Plastic coatings
Paint-tolerant rust preventives
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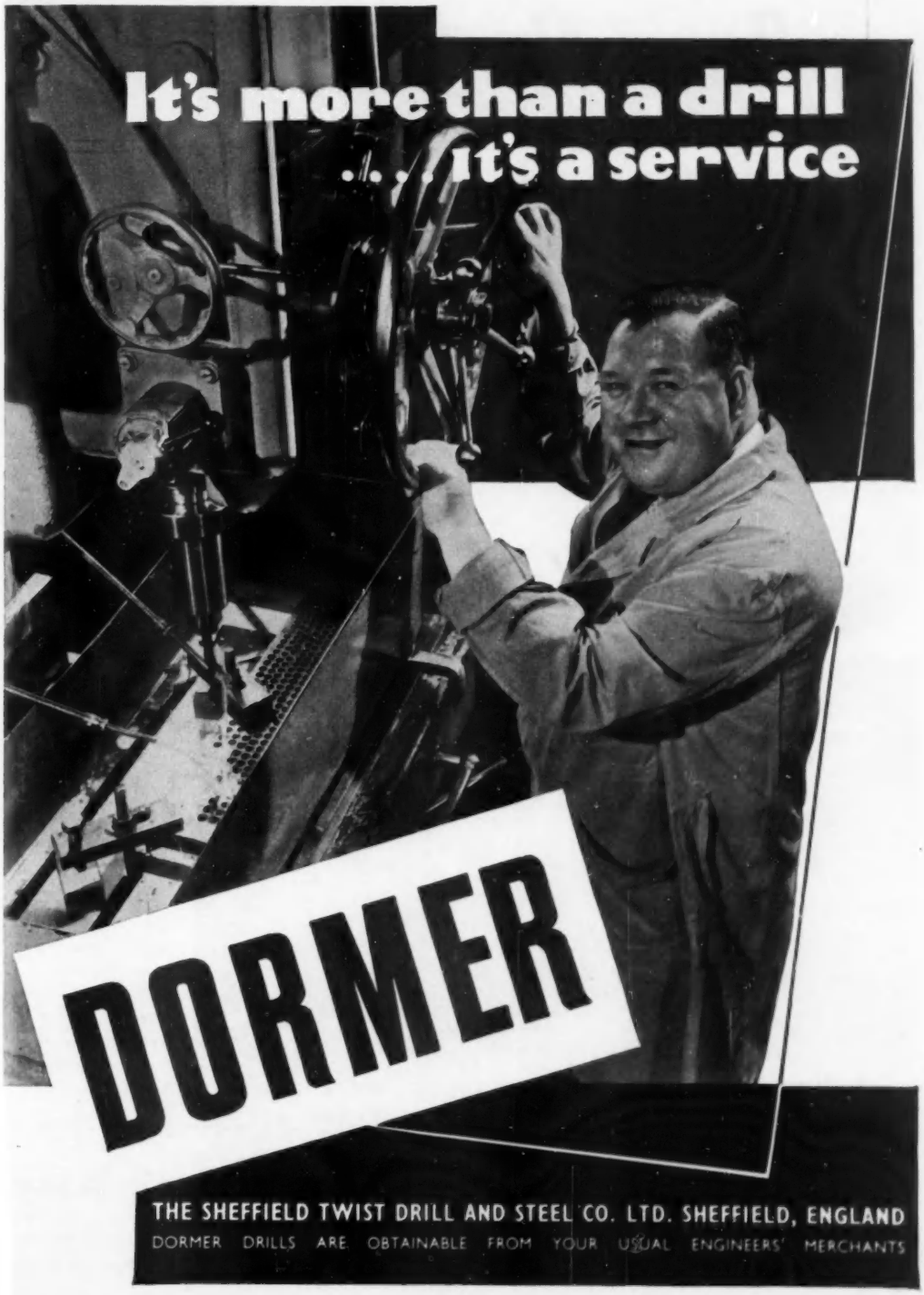
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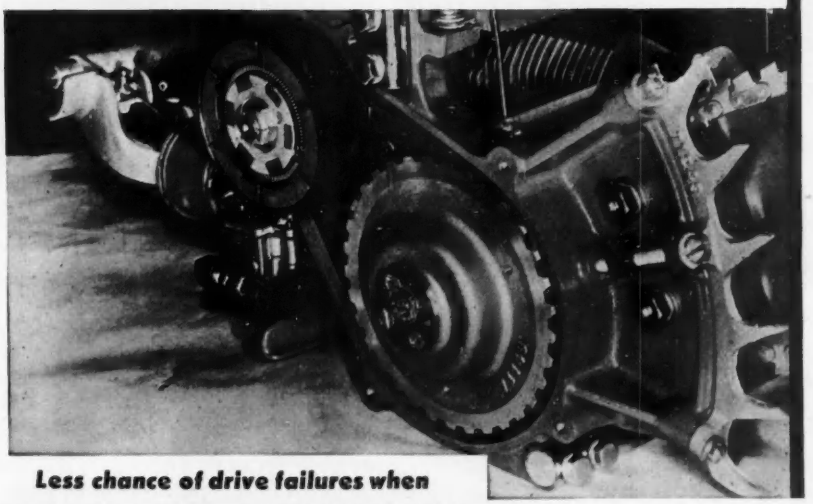

Less chance of drive failures when chain saws rely on POWERGRIP

This chain saw, using a POWERGRIP Timing Belt, withstands up to 700 stop/start cycles per day, with frequent shock stalls. POWERGRIP eliminates slippage and consequent loss of power, while close centres and small pulleys minimize the size and weight of the saw thus making efficient use of power and space available.

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Testing a 40" Hypoid Gear and Pinion for a Pulper, photo by courtesy of Black Clawson International Ltd.

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Spiral Bevel and Hypoid Gears offer many advantages for right angle drives where high efficiency, smooth and silent running are required. E.N.V. specialize in the design and manufacture of straight and curved tooth bevel gears. They offer collaboration with engineers from the project stage on the design of gears and mountings. They have unique facilities for the large-scale manufacture of transmissions including final drives, differentials and axles for vehicles and agricultural equipment, precision gears for aircraft and gears for industrial applications.

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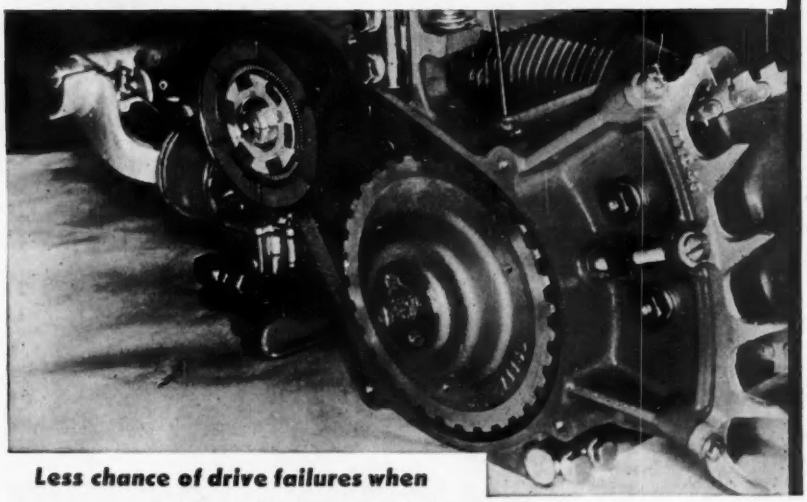

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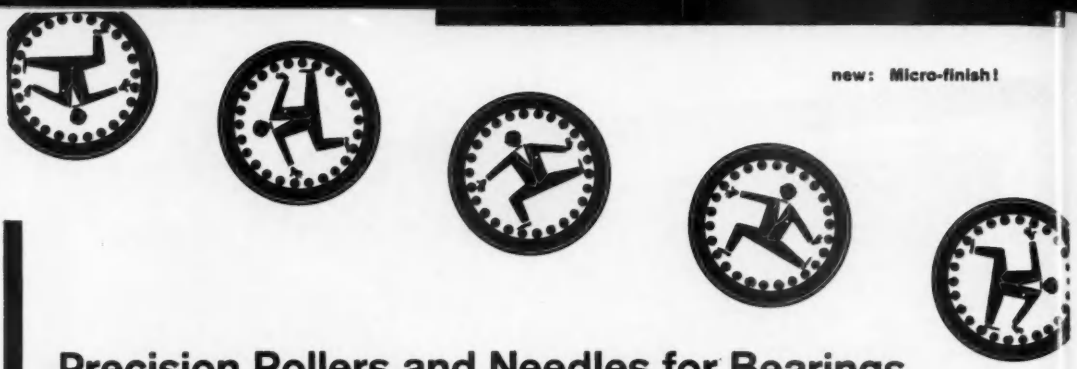

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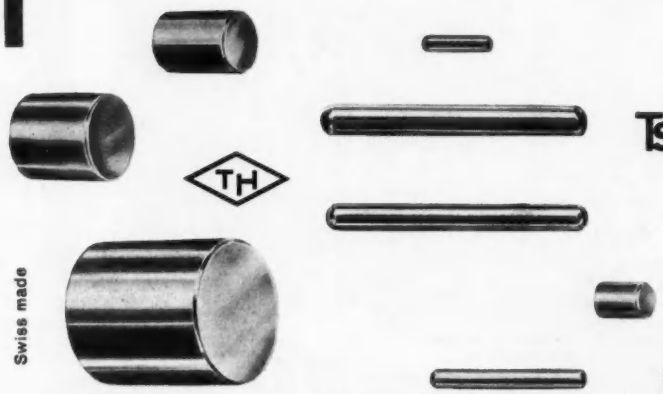

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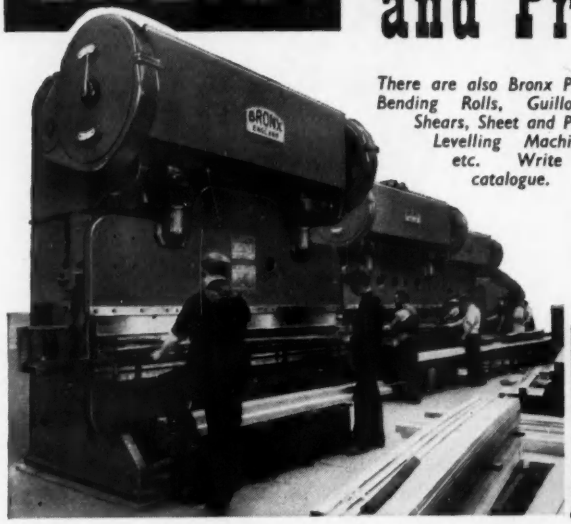

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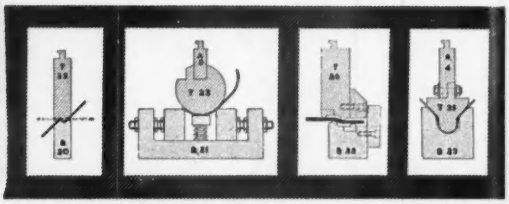

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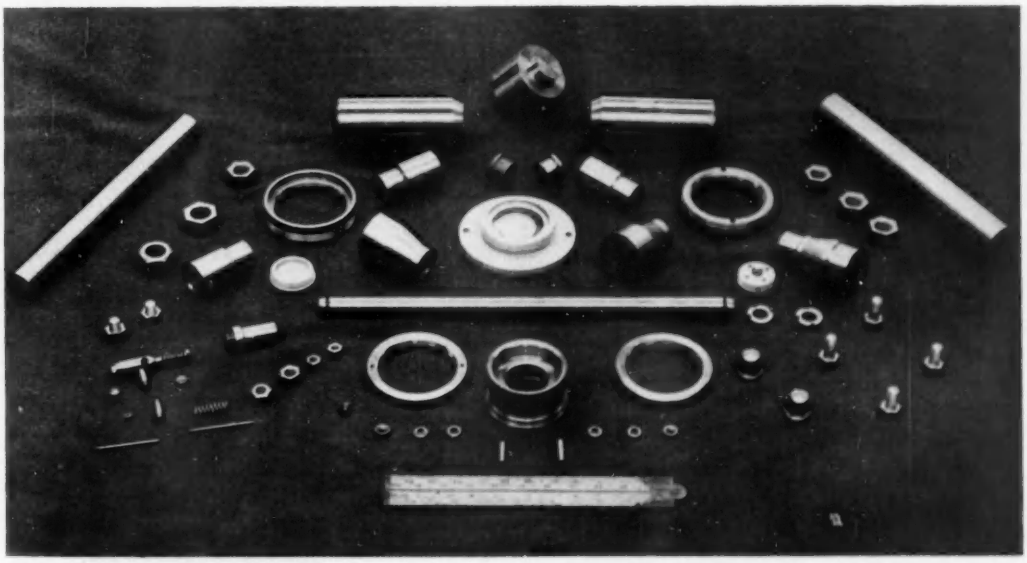

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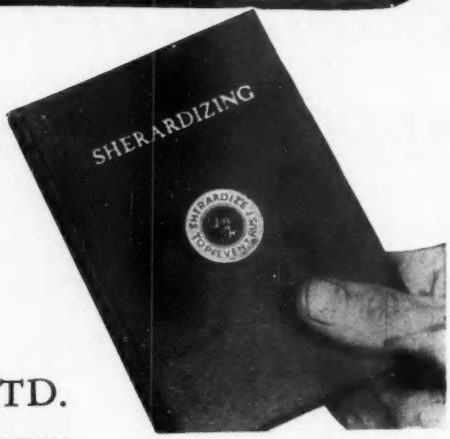

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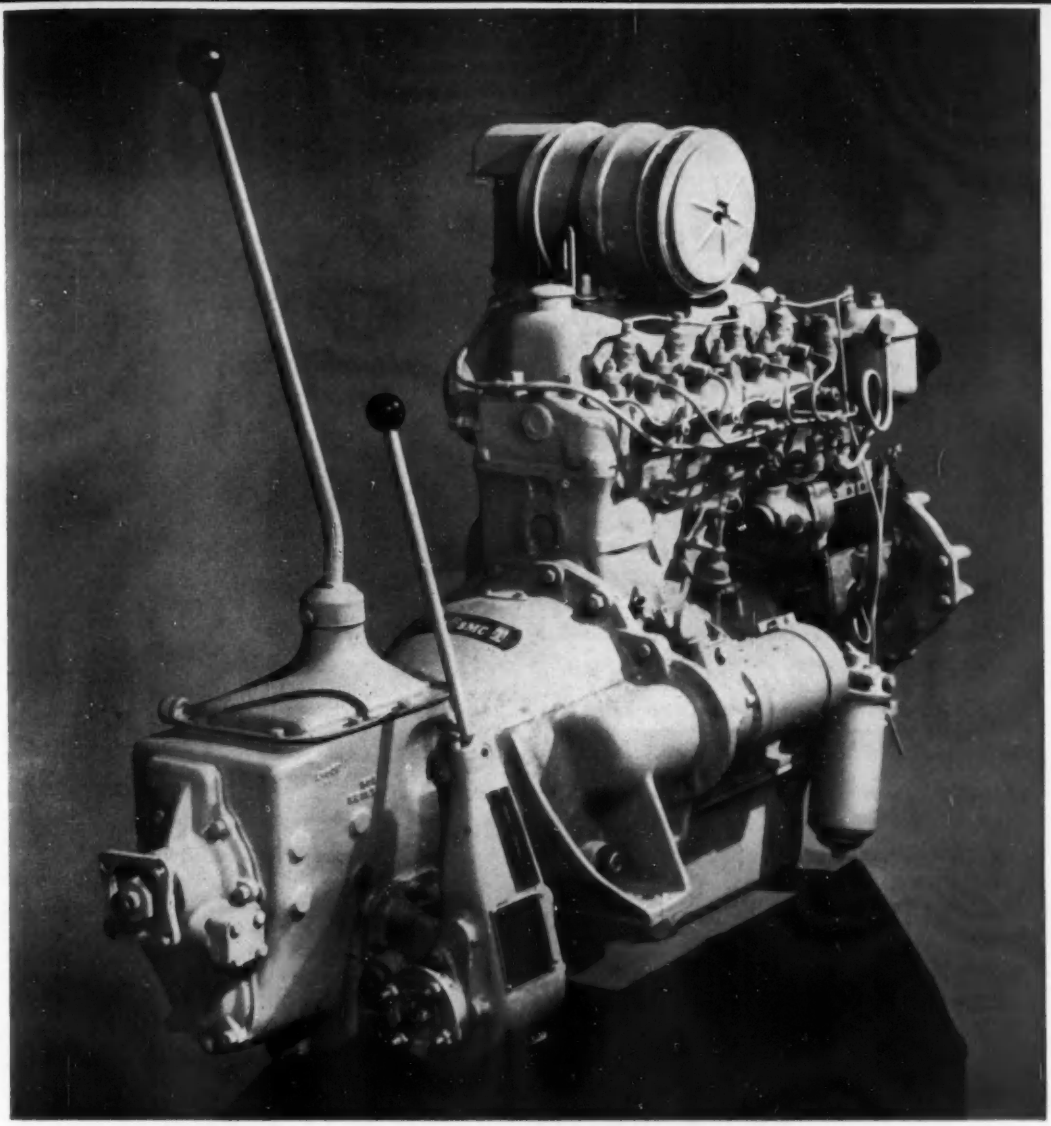

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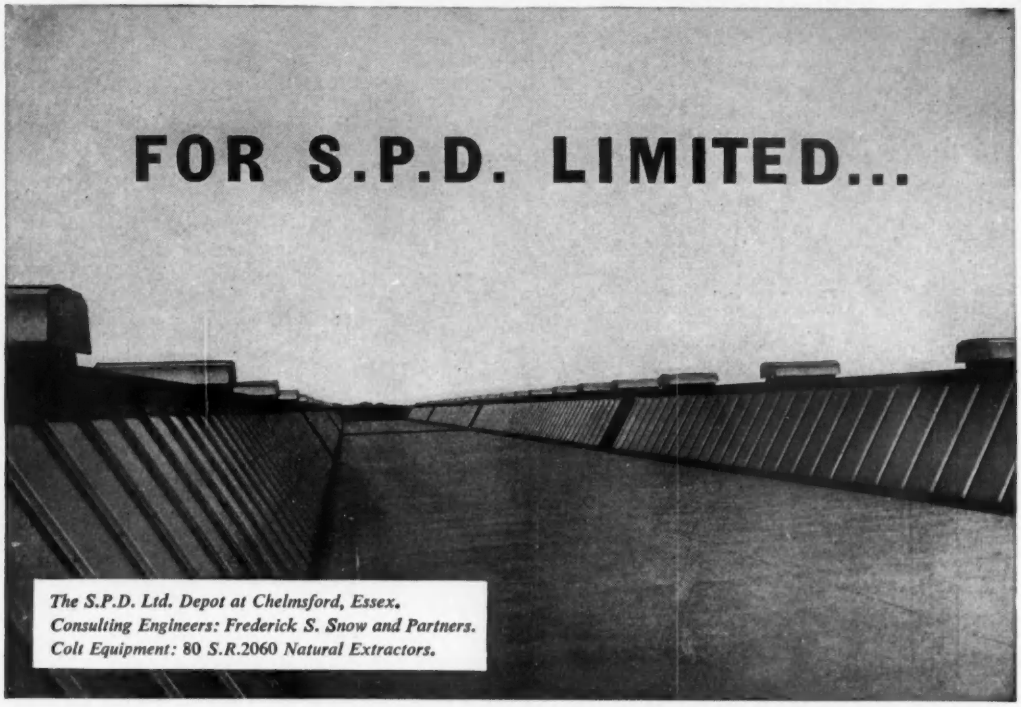

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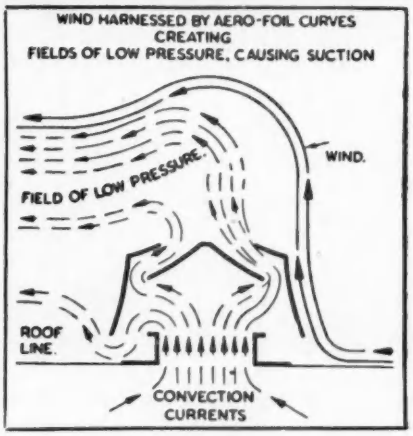

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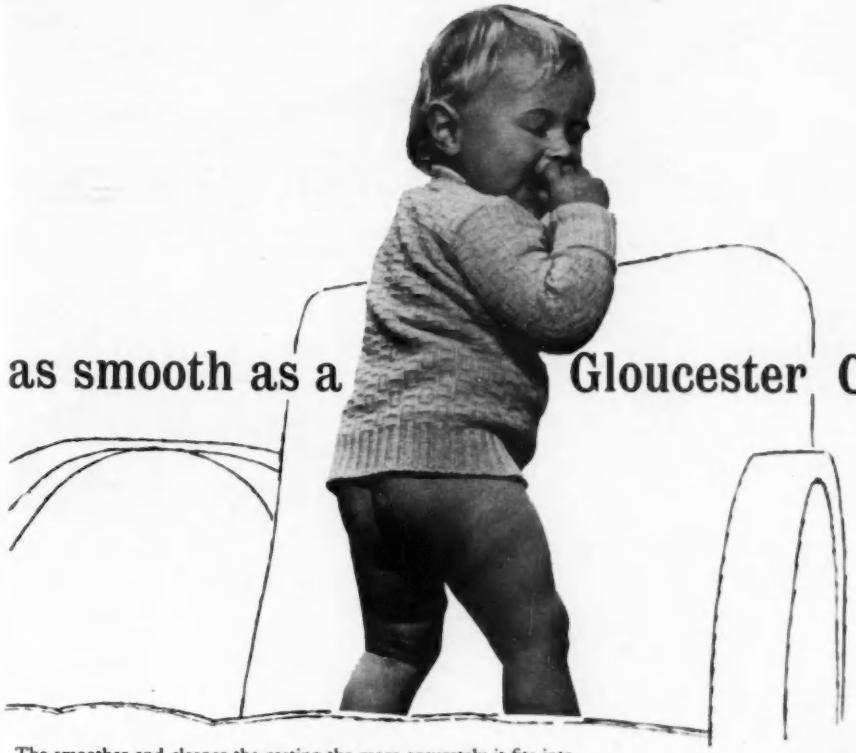

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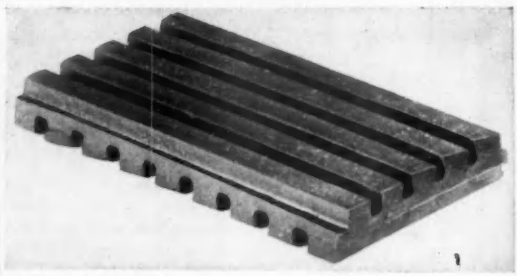

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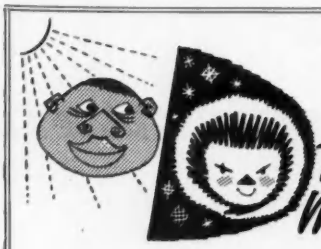

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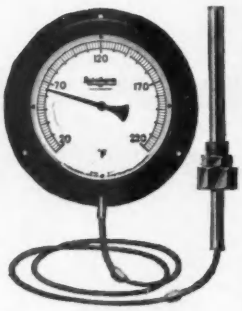

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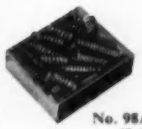

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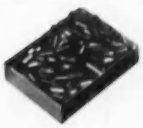

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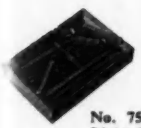

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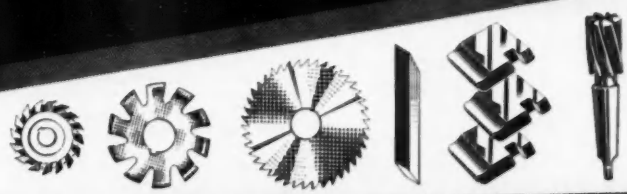

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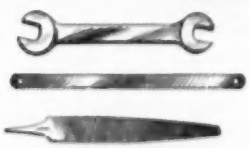

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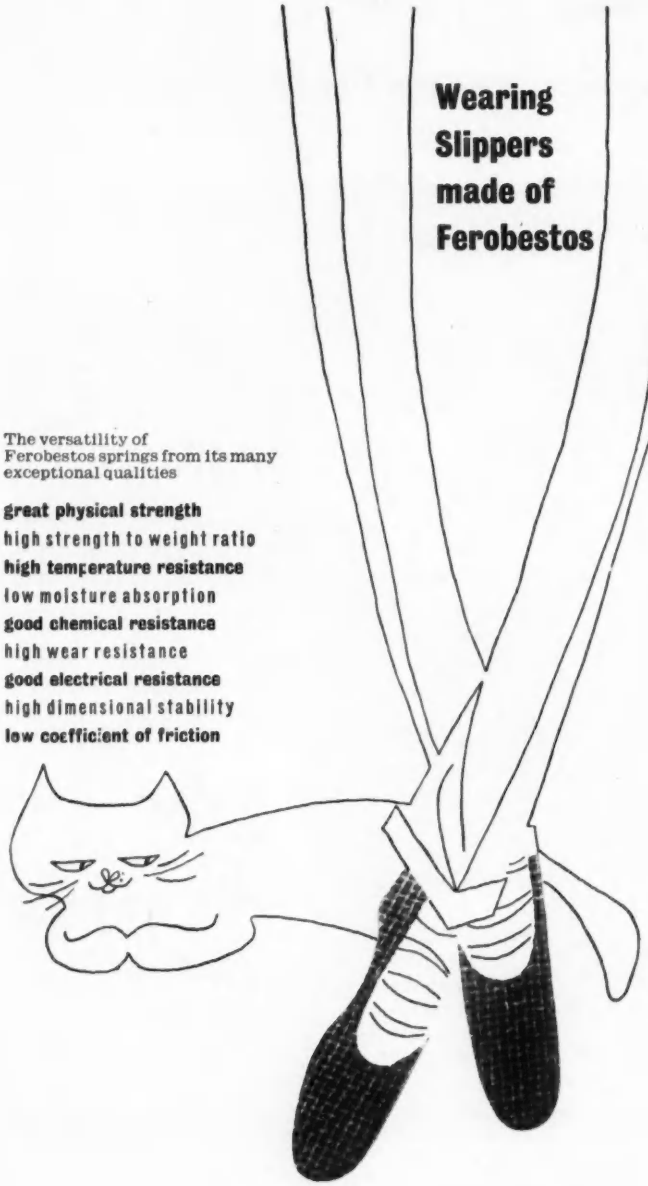

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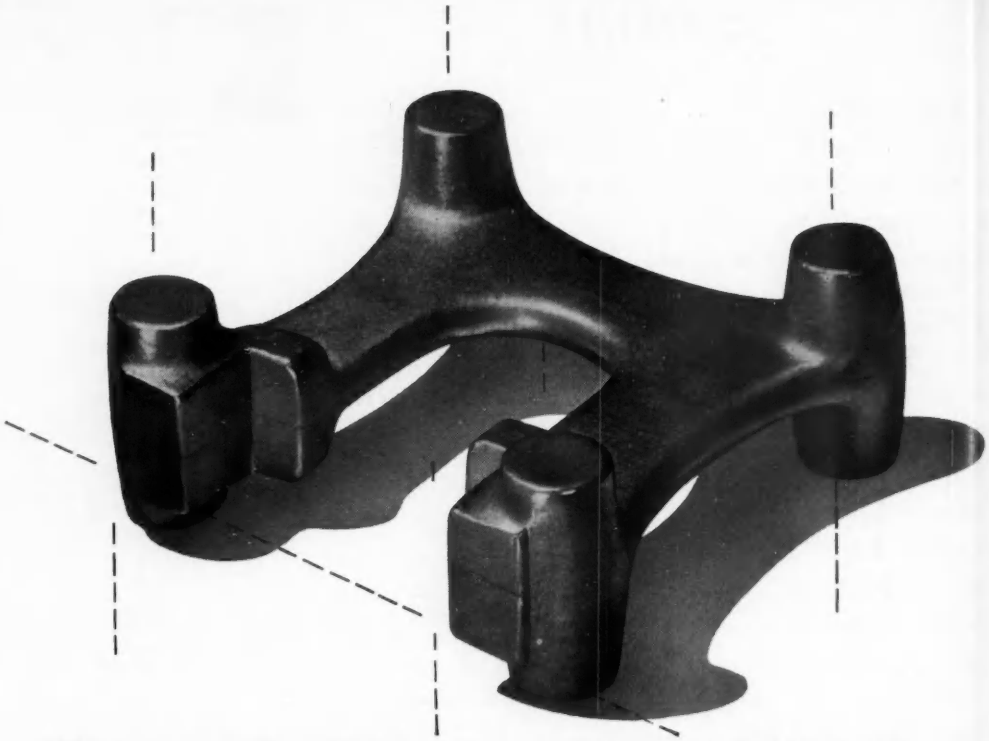

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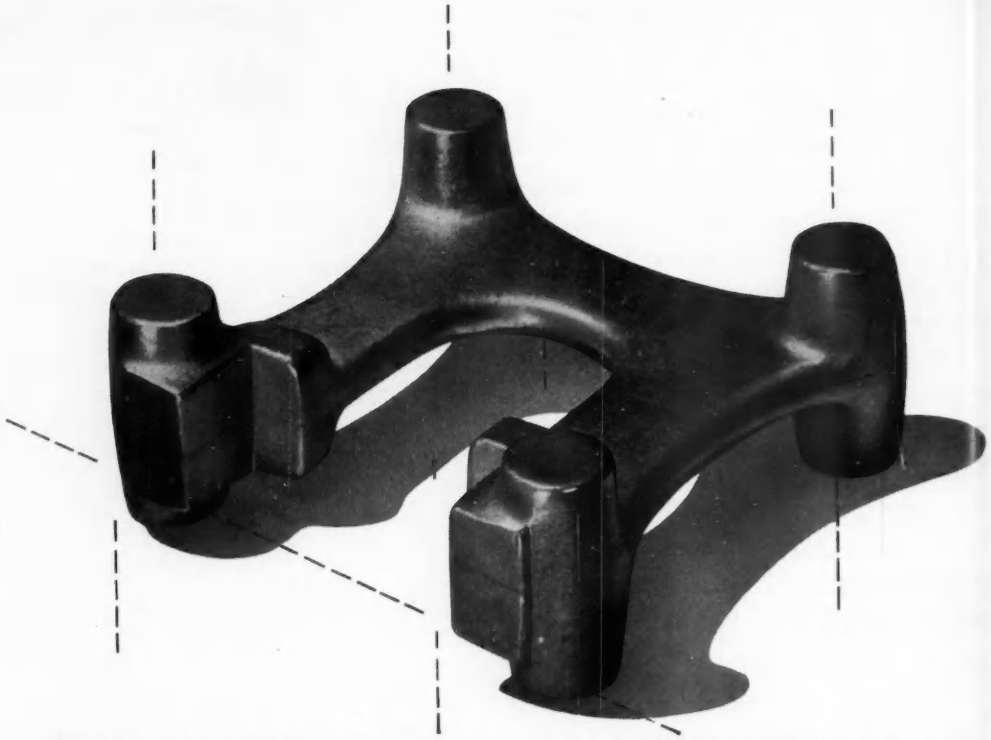
A new production technique for the manufacture of small gas-turbine rotors has been developed and radically reduces manufacturing cost. Essentially, the new technique employs the principle of hot extrusion forging. The individual blades are first arranged in position by means of an outer ring and a central die and, after they have been properly aligned and indexed, the space within the outer ring is filled with molten Kirksite. When this metal has solidified, the central die is removed and replaced by a hot billet. The billet is then extruded under suitable pressure, moving the plastic material under and around the blade roots to effect a secure and completely filled bond, which, it should be noted, is purely mechanical. The extrusion cycle takes place in the fraction of a second and, after extrusion, only relatively minor machining is necessary to remove excess metal from the rotor. It has been ascertained that the natural grain flow inherent in the forging process provides a secure bond of high load-carrying capacity, free from stress concentration. An outstanding feature of the new process is the possibility of producing integral stub shafts or hub extensions on the rotor. Depending upon the size of the extension, a preliminary extrusion is performed on the billet before making the final extrusion. The new process permits the use of any kind of material, including molybdenum, columbium, and titanium alloys, as well as any of the more unconventional alloys. One of the secrets of the process is the use of forging temperatures far beyond those considered to be normally acceptable practice. It is claimed that even the initial attempts proved that the metals, when subjected to high extrusion temperatures, not only retained their original physical properties, but in some instances actually showed improvement.

● *CATHODE-RAY TUBE FOR THREE-DIMENSIONAL INDICATION*

A cathode-ray tube has been developed which makes it possible to present data in true three dimensions. In this tube, which is still in the experimental stage, the fluorescent screen is harmonically displaced with respect to an electron gun, and three-dimensional images are produced by X, Y, and Z gating. For effective operation, persistence of the screen phosphor and duration of the beam gate must be short in comparison with the period of screen oscillation. Also, the screen must oscillate at a frequency equal to or greater than the flicker fusion fre-



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A QUICK GLANCE AT THE LATEST TECHNICAL DEVELOPMENTS THROUGHOUT THE WORLD

● *RADIATION AMPLIFIER OR CONVERTER*

A new composite electro-luminescent screen has been developed, producing on one side of the screen an intensified image of that received on the other side. The screen comprises superimposed layers located between electrodes which transmit the radiation to be amplified or converted and between which an a.c. potential is applied. One of the layers consists of a photo-conductive material dispersed in glass, while a second layer consists of a photo-luminescent material, also dispersed in glass, the two layers being separated by a coherent insulating wafer of a material having a high specific inductive capacity, such as barium titanate or titanium dioxide, which may be in sintered form. The photo-conductive material may be selenium, antimony trisulphide, or cadmium sulphide, while the electro-luminescent material may be zinc sulphide activated with copper or manganese. The conducting electrodes on the external faces of the two layers may be in the form of a metal layer formed by vacuum deposition. As both the photo-conductive and electro-luminescent layers are finely divided materials dispersed in insulation, any variation in the intensity of radiation incident on elementary areas or localized regions of one layer produces corresponding variations in the radiation emitted from similarly located elemenrary areas of the other layer. Thus, if incident radiation falls on the photo-conductive layer through the electrode, localized regions of differing conductivity are produced in this layer, corresponding with the differing intensities of the incident rays. The resulting change in electrical stress excites the correspondingly located regions in the other layer, causing them to be excited and to emit radiation. If the incident radiation on the photo-conductive layer is of a different form from that emitted by the other layer, the device serves as a radiation converter. If, however, the photo-conductive layer is responsive to visible light, then the visible radiation emitted from the electro-luminescent layer represents an amplified image of the incident radiation on the photo-conductive layer.

● *PRODUCTION OF SMALL GAS-TURBINE ROTORS*

A new production technique for the manufacture of small gas-turbine rotors has been developed and radically reduces manufacturing cost. Essentially, the new technique employs the principle of hot extrusion forging. The individual blades are first arranged in position by means of an outer ring and a central die and, after they have been properly aligned and indexed, the space within the outer ring is filled with molten Kirksite. When this metal has solidified, the central die is removed and replaced by a hot billet. The billet is then extruded under suitable pressure, moving the plastic material under and around the blade roots to effect a secure and completely filled bond, which, it should be noted, is purely mechanical. The extrusion cycle takes place in the fraction of a second and, after extrusion, only relatively minor machining is necessary to remove excess metal from the rotor. It has been ascertained that the natural grain flow inherent in the forging process provides a secure bond of high load-carrying capacity, free from stress concentration. An outstanding feature of the new process is the possibility of producing integral stub shafts or hub extensions on the rotor. Depending upon the size of the extension, a preliminary extrusion is performed on the billet before making the final extrusion. The new process permits the use of any kind of material, including molybdenum, columbium, and titanium alloys, as well as any of the more unconventional alloys. One of the secrets of the process is the use of forging temperatures far beyond those considered to be normally acceptable practice. It is claimed that even the initial attempts proved that the metals, when subjected to high extrusion temperatures, not only retained their original physical properties, but in some instances actually showed improvement.

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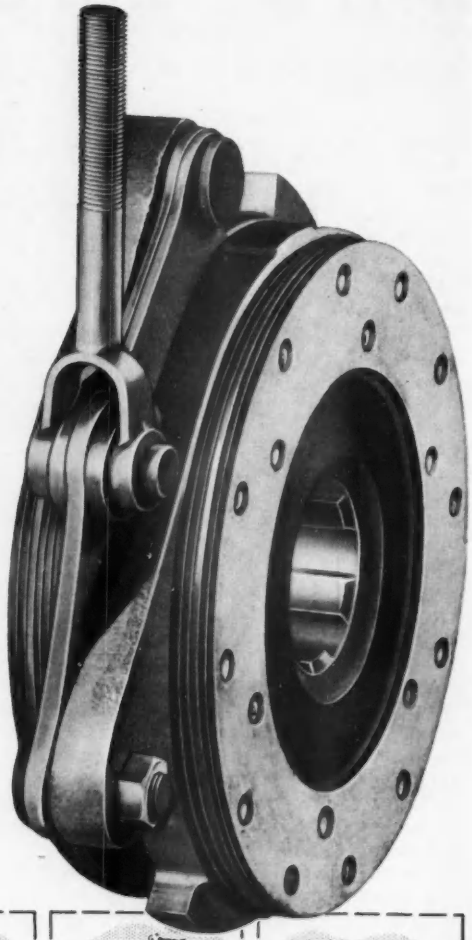

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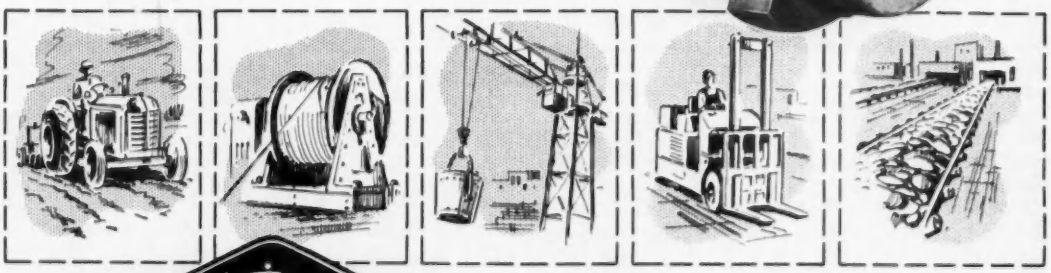

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quency of the eye. Both these conditions are, however, met easily in practice. A large screen can be driven through an appreciable amplitude with comparatively small driving power if the system is in a vacuum, since there is no air load. Large-amplitude oscillation at frequencies up to about 30 cps is facilitated if the mass of the moving structure is kept at a minimum. Since sound is not transmitted in a vacuum, proper acoustic isolation of the drive assembly results in essentially silent operation. The first experimental model of the tube has a screen consisting of a mica disc, 1 mm thick and 18 cm in diameter, coated with a phosphor having a short decay characteristic, light output being limited to essentially the same duration as the beam gate. Crank linkage was selected to give a total screen amplitude of 3 cm. For the 18-cm diameter screen, the XY to Z deflection ratio is 6 : 1. An electron spot can be located anywhere within the total volume of 730 cm^3 swept by the screen per cycle. A typical application of a tube of this kind would be in aircraft traffic control, where it could be used to translate altitude and position data from the airport radar into a three-dimensional display showing both the position and altitude of the aircraft.

● *IMPROVED IRON-ALUMINIUM ALLOYS*

Aluminium added to steel in a fairly high percentage is known to improve the resistance of steel to high-temperature oxidation more effectively than equivalent amounts of chromium. Moreover, it improves the high-temperature strength and decreases the density of such alloys. Steels with 10 to 17% aluminium and 2 to 4% molybdenum have been shown to possess excellent high-temperature oxidation resistance, combined with good stress-rupture and tensile strength at elevated temperatures, as well as good corrosion resistance to many media. Their magnetic properties are similar to those of certain nickel-base alloys, and their electrical resistivity is high. These materials have, however, two drawbacks which have caused many difficulties in fabrication and have thus greatly limited their usefulness. As they are single-phase alloys, grain growth is very rapid at elevated temperature, and ductility at room temperature is very low. However, according to a recent report, the addition of carbon and zirconium in amounts varying between 0·05 to 0·1% and 0·2 to 0·4% respectively has produced startling results, greatly altering the grain-growth characteristics of the material. Investigations have been made to study structural order-disorder variations, so as to determine the effect of this transition on mechanical properties, and it has been found that, by proper heat treatment, optimum overall mechanical properties can be obtained from a structure with a minimum recrystallized grain size and a disordered condition. Minor variations in the carbon and zirconium content and the maximum heat-treatment temperature have been found to have an almost negligible effect on mechanical properties. With its higher ductility, higher tensile strength, and lower ductile-to-brittle transition temperature, the sheet material will be easier to fabricate into usable products, while the effect of zirconium and carbon in stabilizing grain size at elevated temperatures should aid in maintaining a finer grain in the heat-affected zones of a weld.

● *BOLTED JOINT WITH INCREASED SHEAR RESISTANCE*

A new bolt with a spirally knurled body has a head which is similar to an upset rivet head with the head-height of a high-tensile bolt, and is assembled with one nut and washer. Most of the improved performance of the bolt is attributed to the knurling. Each knurl has a ball face which cold-flows grooves in the bolt hole, any notch effect being offset by the cold working. When the bolt is forced into position, excess metal works behind the buttress shape on the rear of the knurls and makes the bolt stick tightly in the hole. The knurls are rolled on the shank in spirals and are relieved to prevent displaced metal from packing, which would make the bolt hard to drive. It is claimed that joints produced in this manner have a shear strength twice that of a riveted joint, and are about 15% better than those produced with standard high-tensile bolts. It is stated that, when used with holes which will accept high-tensile bolts, the new bolts cost no more to install than conventional high-tensile types.

● *ELECTRIC BATTERIES FOR LOW-TEMPERATURE OPERATION*

Electric cells and batteries in general use employ an aqueous solution of some acid or salt as the electrolyte. All these cells and batteries perform under standard conditions of atmospheric pressure and temperature, and most of them can be stored for reasonable periods of time without deterioration. However, the effect of increasing the temperature in storage tends, in general, to reduce their shelf-life, while a reduction in temperature causes them to become inoperative. The chief reason for this characteristic of conventional cells at low temperatures is the type of electrolyte employed, and, while the introduction of certain salts can be used to lower the freezing point of the aqueous solution, it is still impossible to obtain a good cell characteristic at temperatures very much below the freezing point of water. In cases where temperatures greatly below the freezing point of water are encountered, there is no aqueous cell that

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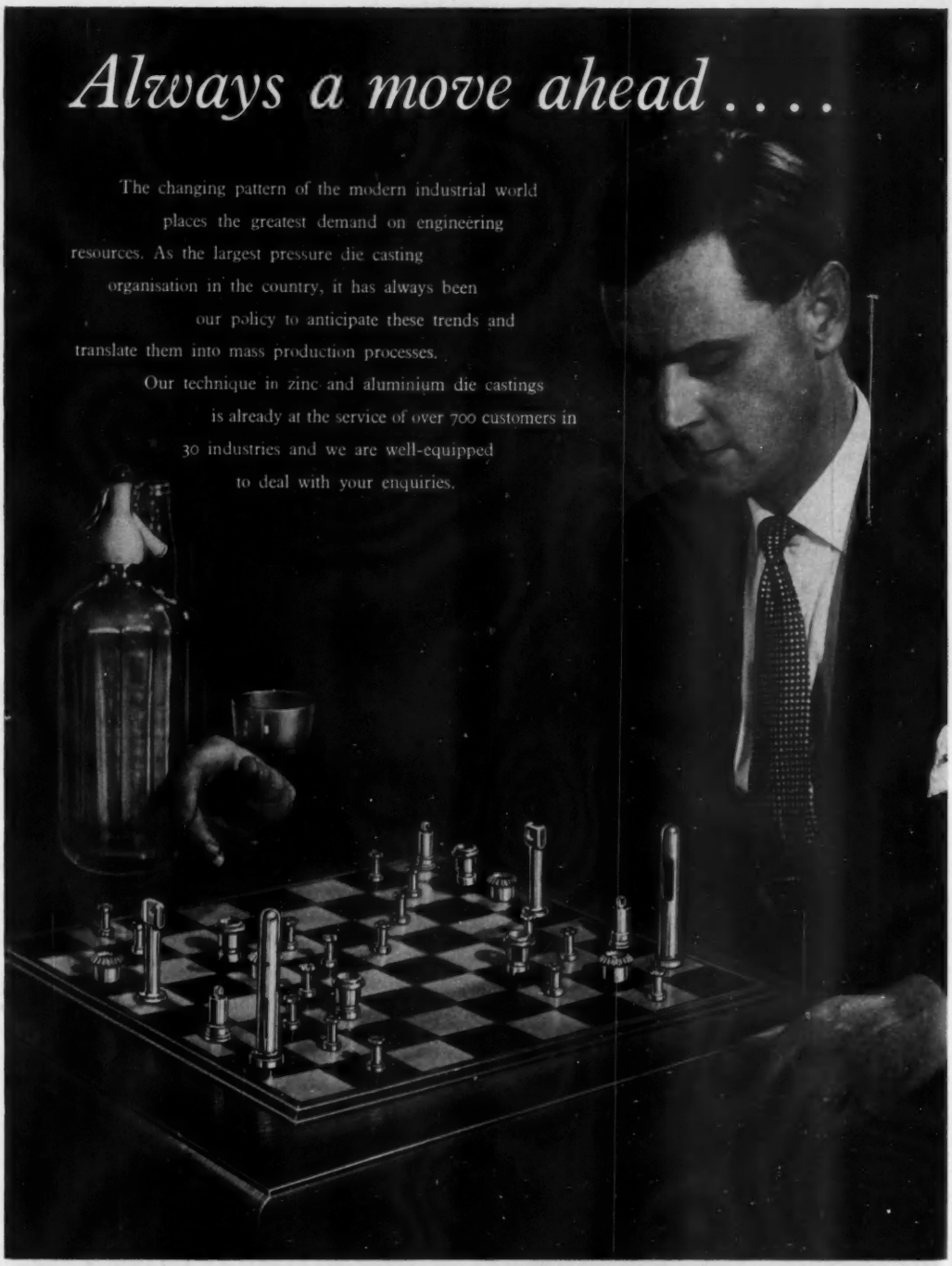

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will operate. These disadvantages are claimed to be overcome in a new type of cell which comprises an anode of an electropositive metal and a cathode, the electrolyte having anhydrous liquid ammonia as a solvent. The electrolyte adjacent to the anode consists of a solution of a salt, the cation of which is a metal corresponding to that of the anode dissolved in anhydrous liquid ammonia ; on the other hand, the electrolyte adjacent to the cathode comprises a solution in anhydrous ammonia of a salt whose cation is ammonium or a metal which develops an electrolyte potential in liquid ammonia of at least 0·75 V less than that developed by the anode in liquid ammonia. It is stated that this type of cell may develop voltages as high as 4 V, as compared with the maximum of 2 V developed by conventional cells, and that it can operate at temperatures ranging from 50°C to considerably below the freezing point of water. In fact, it is stated that the efficiency of the cell is actually improved by reduction of temperature.

● *IMPROVED TUNGSTEN-FILAMENT LAMPS FOR ROUGH SERVICE*

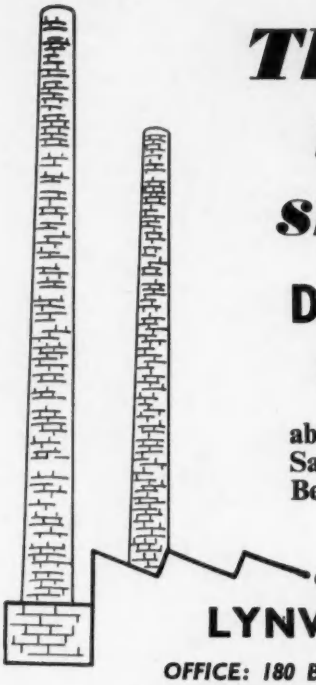
Whilst the majority of the tungsten-filament lamps used for general lighting purposes are nowadays gas-filled, there is still a considerable production of lamps of the vacuum type, especially for use at relatively low wattages, e.g., from 20 to 75 W. Vacuum-type lamps are especially useful under conditions in which they may be subjected to vibration or mechanical shock, as the very fine wire required for a gas-filled lamp of corresponding wattage does not generally stand up so satisfactorily under rough service conditions. A difficulty experienced with such vacuum lamps, however, is that they may in some cases give rise to considerable radio interference, especially in the 30- to 150-Mc/s band. It is believed that this radio interference results from the lamp acting as a Barkhausen-Kurz oscillator by virtue of electrons emitted from the filament which are accelerated towards the glass wall by voltage gradients existing along the filament and then repelled back towards the filament by negative wall charges. In a recent patent specification, however, it is stated that it has been found that radio interference can be largely suppressed by the use of glass bulbs of a size considerably smaller than that hitherto employed for a lamp of the same wattage. It is believed that the reason may be that, at the higher glass-bulb temperatures of the smaller lamp, the resistivity of the path to earth provided by the inner surface of the bulb is considerably reduced and the electronic oscillations damped out, or that the building up of wall charges is reduced. While it is true that reduction of the envelope size of the lamp may well result in bulb blackening and possibly shorter lamp life, it has been found that, by considerably reducing the amount of getter used and ensuring that it is applied only to the filament, it is possible to produce lamps which are little, if at all, inferior to comparable conventional vacuum lamps with respect to bulb blackening and lamp life.

● *BONDING CERMETS TO METALS IN FLUID CONTROL VALVES*

The proper selection of suitable materials for use in fluid control valves operating at high temperatures is extremely important. Operational manipulation, for example, must be amenable to the exacting conditions existing at such elevated temperatures, while the valve-stem and valve-seat materials must possess sufficient rigidity to maintain their shape during service and must be immune to serious solid-phase bonding. At operating temperatures above 1000°F, valves with cermet materials appear to be especially attractive, but the successful incorporation of cermet components into valve assemblies must provide for a method of attaching the cermets to a metallic structural material. A recent report, pointing out that brazing procedures appear to be the most suitable for such cases, covers the development of specific techniques for attaching cermets of the cemented-carbide type to Inconel by brazing. From the research carried out it is concluded that cermets containing large percentages of titanium carbide and tungsten carbide, with possible minor inclusions of solid solutions of carbides of niobium, tantalum, and titanium, can be successfully joined to Inconel. Cermets containing large percentages of tungsten carbide can be wetted with several high-temperature brazing alloys. Of the brazing alloys tested, cermets containing large percentages of titanium carbide were adequately wetted only by alloys of nickel-silicon-boron-iron, nickel-chromium-silicon-boron-iron, and palladium-nickel. A high-temperature direct-bonding procedure has been developed for joining titanium carbide-nickel cermets to nickel, the cermet being placed in intimate contact with the nickel and heated to a very high temperature (approximately 1350°C) in an inert atmosphere.

● *SIMPLIFIED REFRIGERATOR COMPRESSOR*

Small refrigerators operating on the compression principle, especially those of the hermetically sealed type incorporating a thrust piston, are mainly of two designs. In one design the piston is operated by means of a crankshaft, connecting rod, piston pin, and piston, while in the other design, known as the crank-loop type, the connecting rod is replaced by a cylindrical sliding member which reciprocates in a guide bush combined with the piston. The rotary movement of the crank pin is therefore



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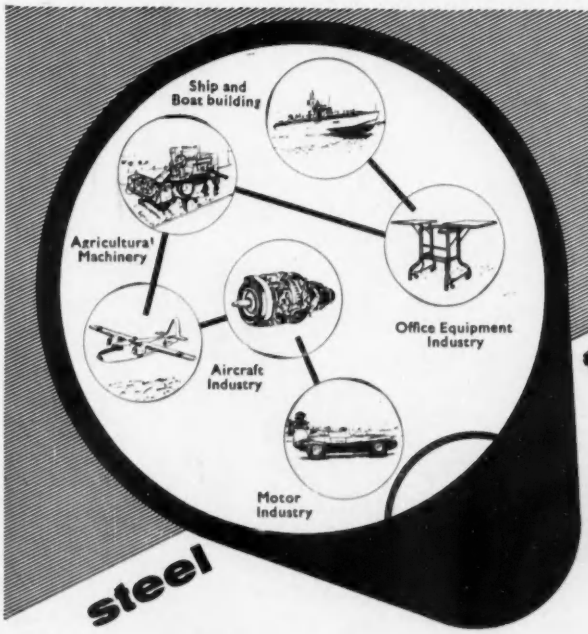

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translated into a reciprocating movement of the piston. Piston operation by means of a crankshaft and connecting rod requires very accurate machining to ensure silent operation and long life of the drive mechanism. The design employing the crank-loop principle is comparatively much simpler, particularly as it requires less accuracy in respect of trueness, since the sliding member is circular, so that the mechanism acts as a universal-joint linkage capable of three-dimensional movement. However, even in this case a number of very accurately machined journals and bores are needed, so that the total of working faces and the amount of friction involved are still quite considerable. A recent patent aims at further simplifying these thrust-piston drives, and at the same time reducing bearing friction and noise of operation. In the drive proposed the piston is reciprocated by a cam disc kept in constant direct frictional engagement with the end surface of the piston. Thus, the rotating cam will slide or roll on the end surface of the piston, constant frictional engagement between piston and cam being ensured by a spring urging the piston against the cam. A further important feature of this design is that the cam is offset with respect to the piston centre, so that the rolling movement of the cam produces rotation of the piston.

● *PRODUCTION OF POWDER-METAL PARTS WITHOUT PRESSING*

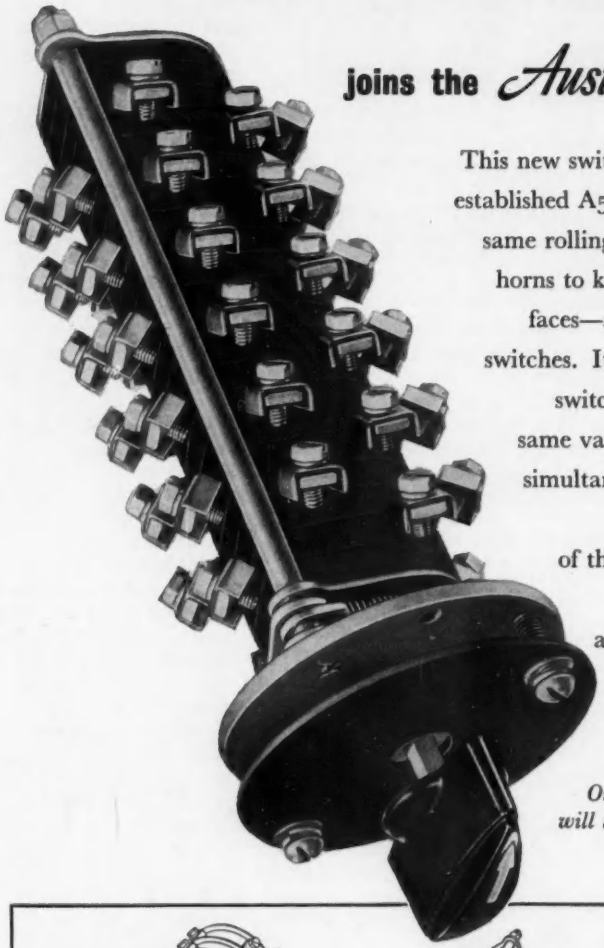
By adapting slip-casting, as employed in the ceramics industry, to powder-metallurgy techniques, it has been found possible to produce hollow parts, especially those which have re-entrant angles or which are too large to be made by conventional powder-pressing. Basically, the slip-casting process consists of preparing a slurry or slip of metal powder, water, and an agent to keep the powder from clumping. This slurry is then poured into a mould made of plaster-of-paris, which then absorbs most of the water. When the mould is broken away, the semi-dry part inside has sufficient green strength to survive careful handling and, after sintering, has the strength and density of a pressed-powder part. Slip-casting is not intended to supersede pressing, but to constitute a complementary process. It does not require expensive presses or dies, and the plaster moulds are of low cost. Compared with powder-pressing, the process is slow, but it appears to be ideally suited to producing small quantities of pieces for which pressing is prohibitively expensive. The solid ingredients of the slip are metals or ceramics ground to about 325 mesh. Stainless steel, silicon nitride, molybdenum, tungsten, chromium, aluminium oxide, and molybdenum disilicide are some of the refractory materials which have been used in the process. About 15% by weight of the slip is distilled water, to which is added less than 1% of a deflocculating agent and binder, such as ammonium alginate, which serves to keep the solid particles in suspension, although most of them are small enough not to be affected by gravity. The acidity or alkalinity of the slip largely determines the density of the final product, so that it may be necessary to adjust the pH value.

● *METHOD OF FORMING COMPOSITES OF ALUMINIUM AND FERROUS METALS*

A recent patent specification describes a new method of forming composite articles consisting of one or more ferrous-metal bodies embedded in and bonded or alloyed to a casing of aluminium or aluminium alloy. Briefly, the method consists of passing an electric current through the iron or steel part, so as to heat it to between 700 and 1100°C, and then casting on the aluminium. The process is carried out by placing the ferrous part in a mould, in which it is suitably supported, and then heating it by the current. The aluminium is poured into the mould in the presence of a de-oxidizing atmosphere, and the heating current is maintained for some time after casting in order to allow sufficient time for the alloying of the ferrous-metal surface with the aluminium to take place. A typical example is in the manufacture of heat-exchanger elements in which a number of stainless-steel tubes are to be embedded in a block of aluminium. The tubes to be embedded are suitably spaced and supported in a mould, the ends of the tubes projecting from the mould. Electrical leads are then connected to the tube ends and the current is switched on. When the tubes have reached a temperature of 700 to 1100°C, molten aluminium is introduced into the mould, using nitrogen as a de-oxidizing atmosphere. In order to ensure filling of the mould cavity, the latter is filled with nitrogen before the introduction of the molten aluminium in the following manner :—The outlet from the mould is sealed with a thin aluminium disc having a small vent hole, whilst the inlet is sealed by an imperforate thin aluminium disc. The nitrogen is then introduced until its escape is detected at the vent hole. Molten aluminium is run into the mould, melting the disc in the inlet and filling the mould cavity. After casting, the heating current is maintained for approximately ten minutes, to cause the metal surface of the tubes to alloy with the aluminium to the required depth.

For further information on developments reviewed in THE BLUE PAGES, readers are invited to write to the Editor.

Now the A15—



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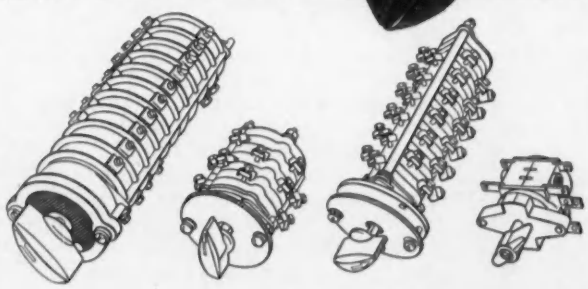

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BY

C. V. Lezard, B.Sc.

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INDUSTRIAL TORQUE CONVERTER

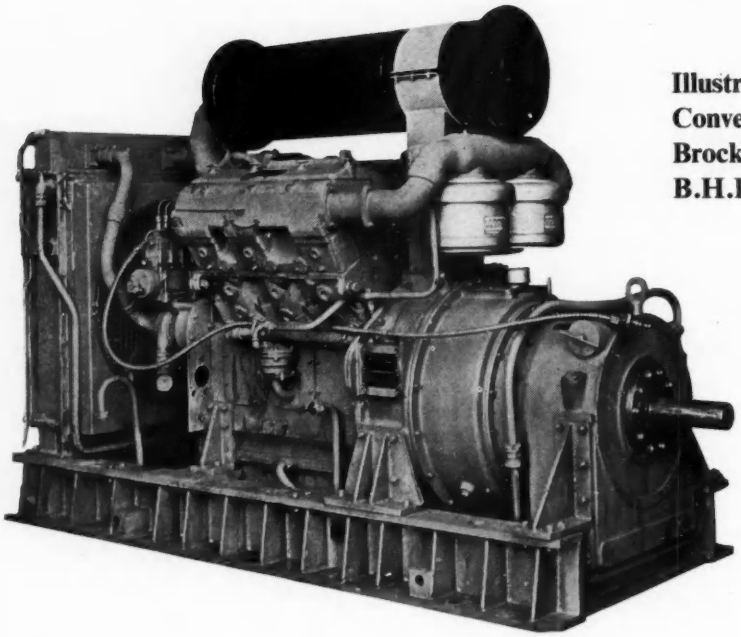

Illustration left shows a Torque Converter Coupling made by Brockhouse fitted to a 200 B.H.P. Paxman engine.

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Hydraulic Variable-Speed Drives

By C. V. Lezard, B.Sc.*

In any application involving an infinitely variable speed transmission system, three alternatives are available, i.e., mechanical, electrical, or hydraulic systems. Previous surveys in this series have dealt with mechanical and electrical variable-speed drives, and this present survey is intended to complete the trilogy by discussing as comprehensively as possible the various types of hydraulic variable-speed systems in major use throughout the world to-day.

The subject is a wide one, and certain omissions have necessarily had to be made. Thus, stepped-speed transmissions, as distinct from infinitely variable types, are not covered, and descriptions are not given of any of the extremely large variety of fixed-displacement pumps produced by many manufacturers, although, as will be shown later, many of them can be used as components of infinitely variable speed drives. Finally, no attempt has been made to summarize the comparative advantages and disadvantages of mechanical, electrical, and hydraulic transmissions, as generalizations regarding their relative suitability for different applications are quite impossible. It may be said, however, that hydraulic units of one type or another are usually best suited to applications in which high shock loads, overloads, frequent speed changes or reversals, and stalling are likely to be encountered. In addition, it is probably true to say that the speed of response of hydraulic systems, including reversing, is considerably greater than that of mechanical and electrical systems, and that, as a rule, they require less space for installation.

Hydrostatic and Hydrokinetic Transmissions

Broadly speaking, all hydraulic transmissions can be divided into two basic types, i.e., hydrostatic (or positive-displacement) and hydrokinetic (or hydrodynamic) types. The hydrostatic transmission comprises a positive-displacement pump unit which delivers fluid under pressure to a similar motor unit, whereas the hydrokinetic transmission fundamentally consists of a centrifugal-type pump or impeller driving a turbine in close proximity. In other words, in a hydrostatic unit, power is transmitted by fluid pressure, without any significant change in velocity, while transmission of energy in a hydrokinetic transmission is effected by a change in velocity of the fluid in the circuit.

Essentially, then, hydrokinetic transmissions are integral units, the impeller and turbine elements being contained in a common casing. Hydrostatic transmissions are also available as package-type units, with the pump and motor units combined in one casing, but they are usually available as split transmissions, i.e., with the pump and motor as separate units. In many applications, separation of the pump and motor units is an advantage, as it permits the motor to be installed at some distance from the pump, with a consequent saving in space, particularly in cramped locations. Also, of course, it is possible to use one pump unit of sufficient output to operate two or more motor units in series or in parallel, according to requirements.

At this stage, it is as well to point out that, of the hydrokinetic units described in this survey, traction-type hydraulic couplings and torque converters are not truly variable-speed transmissions. Thus, it will be noted that details of various makes of traction-type hydraulic couplings (as distinct from scoop-control couplings) are not included, as these are essentially constant-speed units and may be regarded, in fact, as hydraulic clutches. The objection may therefore be raised that torque converters, and converter-couplings should similarly be excluded, inasmuch as (with few exceptions) they also do not incorporate manual or other provision for adjustment of output speed. The inclusion of these latter units is, however, considered justified, inasmuch as they have what may be called an automatic variable-speed feature, output speed being automatically adjusted in accordance with torque requirements.

In this connection, it is interesting to note that, in the U.S.A. at least, a distinction is made regarding types of variable-speed transmissions. Thus, a transmission in which provision is made for varying, as required, the output speed, e.g., in a variable-delivery pump and motor combination, is designated an adjustable-speed drive, while a transmission in which speed variations cannot be controlled arbitrarily by human or other intervention, e.g., a conventional torque converter, is designated a variable-speed drive.

Hydraulic Fluids

Nearly all hydraulic transmissions utilize oil as the fluid medium, and only the highest grades should be used. In short, manufacturers' recommendations should invariably be adhered to, particularly with regard to viscosity and operational temperatures. In addition, bearing in mind the high-precision construction of many hydraulic-transmission components, e.g., pumps, scrupulous cleanliness of the oil and, indeed, of all parts of the circuit is essential.

Information Required when Contemplating Hydraulic Transmissions

Although indications are given in the following pages of the capacities of most of the hydraulic transmissions described, it should be appreciated that such indications can only be interpreted broadly and are intended merely as a rough guide. The performance characteristics of an infinitely variable speed hydraulic unit depend on a number of factors, necessitating close co-operation between prospective purchasers and manufacturers of such equipment.

As a general guide, the following information may be considered necessary to form a basis for the selection of appropriate hydraulic transmission equipment, particularly insofar as hydrostatic types are concerned :—

(1) Type of machine to be driven.
(2) Maximum and minimum output speeds required and direction of rotation.
(3) Type, power, speed, and direction of rotation of drive motor.
(4) Maximum horsepower and torque to be transmitted.

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(5) Nature of load, i.e., steady or fluctuating.
(6) Requirements regarding constant-horsepower and/or constant-torque output.
(7) Operational conditions, i.e., ambient temperature, space available, normal running time per day, and approximate running time at various speeds.
(8) Requirements regarding full reversibility or otherwise of the driven machine.
(9) Preferred method of speed control, i.e., manual, electrical, hydraulic, etc.
(10) Proposed method of coupling hydraulic units to the drive motor and driven machine.

HYDROSTATIC DRIVES

As already mentioned, hydrostatic transmissions basically comprise a positive-displacement pump delivering fluid under pressure to a corresponding hydraulic motor. Rotary types of units are used almost exclusively and, for the purpose of this survey, have been divided into various groups, i.e., plunger types, with radially or axially disposed pistons, vane types, and special types, all of which are fitted with means for variable displacement of either the pump unit, the motor unit, or both.

As a result of limiting descriptions to units with a variable-displacement feature, details of a large number of fixed-displacement pumps, including gear pumps and screw pumps, which are produced by many manufacturers throughout the world, have been omitted. Nevertheless, fixed-displacement pumps of these well-known types, as well as other types, can be, and very often are, used in conjunction with fixed-displacement motors of different types to produce, in combination with suitable valving arrangements, infinitely variable speed hydraulic transmissions, and some brief details of these arrangements are given below.

Speed Variation with Fixed-Displacement Pumps and Motors

A combination of a fixed-displacement pump and motor can provide infinitely variable speed regulation if a throttling valve, which may be a needle, gate, or other type acting as a variable restriction, is included in series in the circuit. In throttling circuits of this type, which are extremely economical, insofar as first cost is concerned, some of the output from the pump must be by-passed or spilled through a relief valve. Consequently, such circuits must operate at relief-valve pressure if any speed control is to be obtained, so that power input to the pump is constant, irrespective of power output. The efficiency of this system can be expressed in terms of the ratio of output horsepower to input horsepower and, as can readily be appreciated, can be very poor at speeds and torques considerably less than maximum, as compared with a system using a pump of identical characteristics, but with a variable-delivery feature and no throttling valve in the circuit. Fig. 1 gives comparative indications of efficiency as a function of output speed, assuming constant output torque, for a system using a fixed-displacement motor and pump, with a relief valve and a throttling valve for speed variations, and an identical motor and pump, the latter, however, with a variable-delivery feature and therefore no throttling valve in the circuit.

As a result of their poor efficiency at low speeds, it is obvious that circuits of this type are attractive only when operation is mainly at maximum speed and when torque loads on the motor are more or less constant. At the same time, however, it should always be borne in mind that a variable-displacement pump or motor is appreciably more expensive than a fixed-displacement type, so that, particularly where low-power applications are concerned, it is desirable to take into consideration whether the additional cost of operation, in terms of electrical consumption, engendered by poor efficiency at low speeds, is not offset by the difference in cost between a variable- and a fixed-displacement unit.

It is also possible to provide infinitely variable speed control by means of a throttling valve in parallel with the motor. In these systems, usually called by-pass circuits, motor speed can be reduced without by-passing oil through the relief valve, and their efficiency is the ratio of actual speed to maximum speed, regardless of output torque.

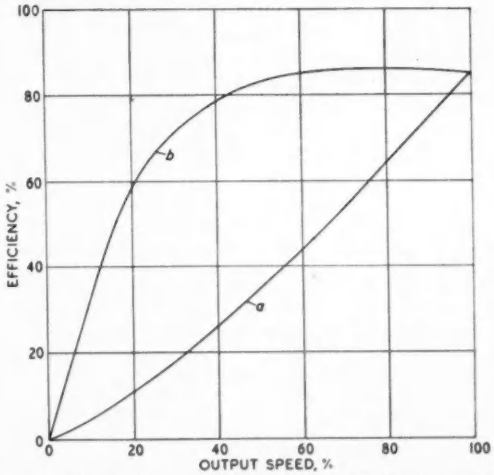


Fig. 1. Comparative curves of efficiency as a function of output speed, assuming constant output torque, of a fixed-displacement motor and pump combination with a throttling valve in the circuit for speed variation (curve *a*), and an identical motor and variable-delivery pump without throttling valve (curve *b*).

By using two fixed-displacement pumps of different volumes and appropriate valving arrangements, stepped speed control of a fixed-displacement motor can be obtained, providing three different output speeds by using one or the other or both of the pump units. By incorporating throttling or by-pass circuits, infinitely variable speed control of each step is possible, thereby considerably reducing spillage losses. Double pumps for this purpose, with a common housing, shaft, and inlet port, are available, but the use of more than two fixed-displacement pumps is usually not practicable for economic

reasons, as their cost, together with the corresponding valving and controls, could easily exceed that of a variable-delivery unit.

In systems of this kind, reversal of the motor shaft is easily accomplished by a suitable four-way valve, some types of which permit them to be used also as flow-control valves. In this connection, a wide variety of valving equipment of different types and for different applications is produced by numerous manufacturers throughout the world, including Racine Hydraulics & Machinery Inc., in the U.S.A., who also produce a range of pumps (described later in the section on vane pumps) which incorporate a variable-volume pressure-compensating control which automatically maintains a preset pressure and eliminates the need for any relief valve in the circuit, thus avoiding by-passing and reducing power consumption by strictly limiting delivery to output demands.

The above by no means exhausts the different types of circuits and valving arrangements possible. In fact, it may be said that the ability to create variations or combinations of several basic arrangements for specific applications by means of proper selection of hydraulic valving and circuits is an inherent advantage of variable-speed hydraulic transmissions of this type.

There are, of course, many manufacturers of fixed-displacement pumps and motors and, indeed, most of the firms who produce variable-delivery pumps and motors of the types described in the following also produce fixed-delivery versions. Some firms, in fact, specialize in designing hydraulic infinitely variable speed drives incorporating fixed-displacement pumps and motors, with the appropriate valving and circuit arrangements for different applications.

Pump and Motor Characteristics

Neglecting efficiency, the horsepower required by a pump in a given application, or the power output of a hydraulic motor, can be estimated very closely from the following relations :

$$\text{HP} = \frac{pQ}{1430} = \frac{nT}{63{,}000}$$

where p = pressure [psi], Q = delivery [Imp. gal/min], n = speed [rpm], and T = torque [lb-in.].

From the above it will be apparent that horsepower is in linear proportion to both pressure and delivery, which, however, itself depends on displacement (i.e., on bore and/or stroke) and on speed. Thus, at a given pressure p, if the delivery Q is kept constant, two motors of equal rating can be designed for operation at different speeds by increasing or decreasing their displacement. For example, in a low-speed application, the bore and/or stroke of the pistons can be increased, while in a high-speed application, they can be reduced.

As already stated, horsepower input is also proportional to system pressure, which itself depends on load requirements, i.e., torque, at a given speed. Consequently, if load requirements are high, all components in the system must be capable of withstanding the high pressures involved, and the input power must similarly be increased. It is for this reason that many manufacturers of hydraulic pumps and motors specify performance data in terms of several operational pressures, indicating that these units are capable of functioning safely at the maximum pressure stipulated, but that they can also be used at lower pressures with correspondingly decreased ratings.

In general, it may be said that vane-type units are suitable for operation up to about 2200 rpm for units of small power rating, i.e., up to about 5 hp, and up to about 1200 rpm for units up to about 50 hp, with recommended minimum speeds of approximately 100 rpm for motors and 600 rpm for pumps.

Where torque, permissible operating pressures, and efficiency (70 to 80%) are adequate for a given application, the vane-type unit undoubtedly shows a saving in cost over most piston-type units. Where, however, higher torque, operating pressures, and efficiency (80 to 90%) are considered desirable, a combination of radial-piston or axial-piston pump and motor is normally preferable. With piston-type units, full-load torque can be delivered at very low speeds. Radial-piston units are capable of delivering appreciably greater powers at a reasonable speed, e.g., up to 750 hp at 1500 rpm, than is possible with vane-type units, while axial-piston units can also be supplied for high power ratings and are usually capable of operation at higher speeds than radial-piston units, in some cases up to nearly 10,000 rpm. On the other hand, for very low speed applications, e.g., 50 rpm maximum, the radial-piston motor is normally used.

Insofar as the overall efficiency of hydraulic units is concerned, it is true that most manufacturers include curves of efficiency in their sales literature ; however, it is felt that the reproduction of such curves in this survey might well be misleading, as the curves from different manufacturers would yield truly valid comparisons only if they were plotted under basically similar conditions of operation. In this connection, it must be appreciated that the overall efficiency of hydraulic transmissions depends on a number of factors, including input speed, output speed and range, system pressure, torque requirements, viscosity of hydraulic fluid and ambient temperature, length of piping between motor and pump units and its dimensions, and flexibility of control, i.e., variable-delivery pump or variable-displacement motor, or both variable.

Finally, it should be noted that hydraulic pumps and motors normally require a high degree of manufacturing accuracy, the idea being to reduce inevitable leakage losses to a minimum. While such factors as friction in pistons, bearings, and sliding components, flow losses, compression losses, etc. all play their part in determining the overall efficiency of a hydraulic transmission, leakage losses are the major individual factor, being most pronounced at full load. In short, the whole picture of efficiency is a very complicated one, depending not only on the inherent design characteristics of the units concerned but also on how they are installed and operated.

Radial-Piston Types

As their name implies, radial-piston types of hydraulic pumps and motors utilize a cylinder block in which an odd number of pistons is disposed radially, variable delivery, i.e., variable piston stroke, being obtained either by the use of an eccentric or cam on the drive shaft or of an exterior drum or reaction ring, the eccentricity of which can be altered, as required. The outer ends of the pistons are kept in contact with the surface of the reaction rings by centrifugal force, which may be augmented by springs or by pressure from a booster pump. These outer ends can transmit reaction to the ring by various methods, e.g., segmental shoes or " slippers ", bevelled reaction rings, or rollers. Generally, whatever method is used to transmit reaction to the ring, the rotor revolves about a pintle (Fig. 2), which fulfils the function

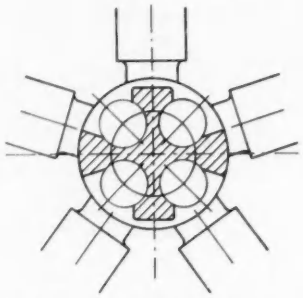


Fig. 2. Section of a typical pintle and cylinder rotor.

of a distribution valve for discharge and suction. To allow the outer ends of the pistons to operate in a bath of oil, the adjacent area is sealed by forming the rotor halves into a casing, or by the actual provision of a casing between them. Lubrication is provided by the slippage oil from the pistons accumulating in this space.

An example of a radial-piston design using segmental slippers is the Hele-Shaw variable-delivery hydraulic pump. Originally intended for use in World War I for the hydraulic operation of marine auxiliaries, including steering gear, this British design, manufactured in the U.K. by MacTaggart Scott & Co. Ltd., and in the U.S.A. by the American Engineering Co., may be said to be the doyen of radial-piston units.

In the Hele-Shaw pump (Fig. 3), the rotating cylinder body carries the pistons (1), their gudgeon pins (2), and the slippers (3) around with it, the slippers in their turn causing the floating ring (4) to rotate on rolling bearings. The slippers, whose resistance in their annular groove is greater than that of the bearings, approach and recede from each other in each revolution by an amount equal to the stroke of the pump. The floating ring (4) is made in the form of a drum, and oil leaking past the pistons (1) is retained by centrifugal force as an annulus to lubricate the slippers (3) and gudgeon pins (2). As all forces in the pump are in the plane of rotation, there is no end thrust, and a small end float can be permitted. The cylinder body, which is coupled to and driven directly by the primer mover rotates on a fixed centre valve or pintle (5), in which are located the suction and delivery ports, communicating with the exterior by passages.

The oil required for lubricating the internal mechanism of the pump is derived from the pump itself, in the form of the small leakage past the pistons and, after performing its duty as a lubricant, falls to the bottom of the pump casing, where holes are provided for its escape. A tank. The stroke of the pump can easily be varied from receiving this oil, and some form of pump, e.g., a small gear pump, which can be driven from the main pump shaft, installed for returning the oil to the main supply tank. The stroke of the pump can easily be varied from the neutral or no-stroke position to the full-stroke or full-delivery position while the pump is operating under full-load conditions ; furthermore, the direction of flow of the oil from the pump is reversible, merely by moving a slide to the right or to the left of the neutral position.

These Hele-Shaw variable-delivery pumps are available in a range of sizes with outputs for continuous duty from about 2½ hp at 1000 rpm and 1500 psi to about 60 hp at 480 rpm and 1500 psi. In general, this design of

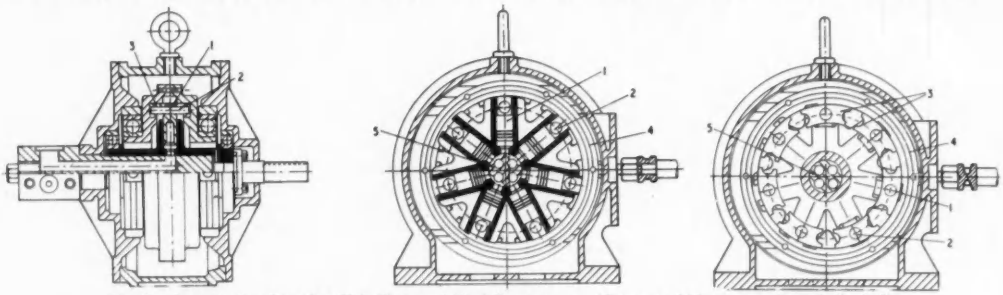


Fig. 3. Cross-section through a Hele-Shaw variable-delivery pump. (*Courtesy of MacTaggart Scott & Co. Ltd.*)

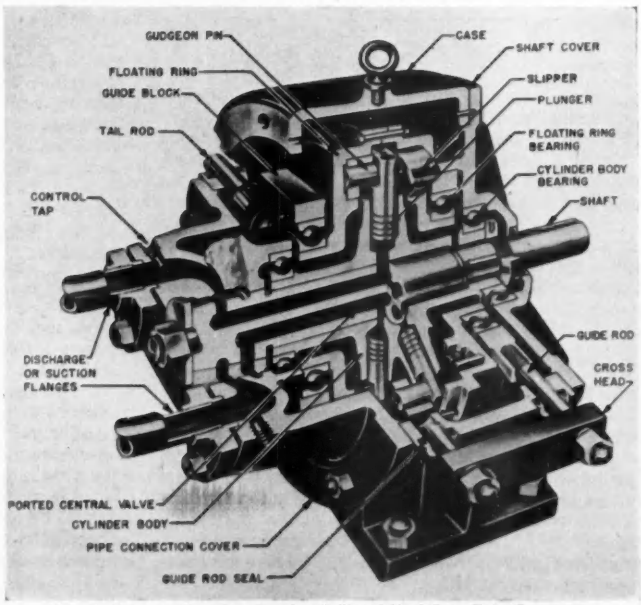


Fig. 4. Cut-away view of an American-built variable-delivery Hele-Shaw pump. (*Courtesy of American Engineering Co.*)

pump is not suitable for use as a hydraulic motor, because of its low starting torque. Consequently, a range of special motor units has been developed. These units, which are of the cam type, are available in several sizes, including special low-speed models with capacities ranging from about 18 to 90 hp at 50 rpm, and three-, five-, and seven-cylinder units with capacities from about 10 hp at 500 rpm to 30 hp at 500 rpm.

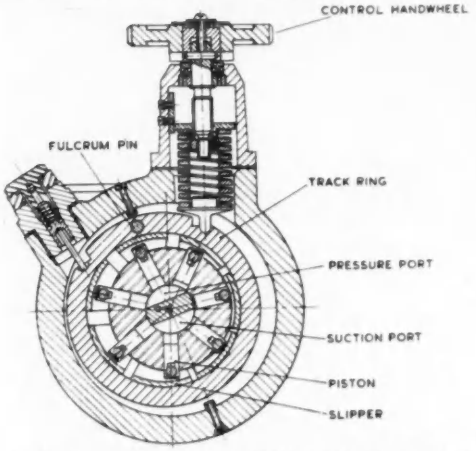


Fig. 5. **Cross-section through an "HP" model A.V.D. variable-delivery pump.**
(Courtesy of Hydraulics & Pneumatics Ltd.)

A cut-away view of a slipper-type Hele-Shaw variable-delivery pump made in the U.S.A. by the American Engineering Co. is shown in Fig. 4. These pumps, which can also be used as hydraulic motors, are available in a wide range of high-pressure and low-pressure versions, with capacities from about 1 hp at 1150 rpm up to about 80 hp at 675 rpm.

The range of radial-piston variable-delivery pumps produced in the U.S.A. by the Hydraulic Power Division of The Hydraulic Press Mfg. Company, utilizes pistons which are one-piece, T-shaped, alloy-steel forgings, with the crossheads fitted in T-shaped thrust-shoes, which are clamped between the two halves of the reaction rotor. These units are available in various sizes, with capacities ranging from about 2 hp at 1200 rpm and 2500 psi up to about 300 hp at 720 rpm and 2500 psi.

Another design of radial pump employing slippers pressing against an eccentrically mounted reaction ring is that produced in the U.K. by Hydraulics & Pneumatics Ltd. Although this unit (Fig. 5) is fundamentally intended for use as a pump, with a maximum input rating of about 2½ hp at 1500 rpm and 2500 psi, it can also be used as a hydraulic motor, so that a combination of pump and motor can be used to produce a hydraulic variable-speed drive.

An example of a design using an eccentric is the Hele-Shaw Beacham variable-delivery hydraulic pump, made in the U.K. by Greenwood & Batley Ltd. In this design (Fig. 6), the pump comprises a cylinder block (1), with a number of radial cylinders rotating in a casing (2), to which is secured a crank arrangement consisting of two steel eccentrics (3) and (4), which fit one inside the other and which have an eccentricity that is variable from a maximum at one side of the centre of rotation of the cylinder block to a similar maximum at the opposite side of the centre, the crank passing through a position of zero throw at the actual centre of rotation. The pistons are connected to the crank by connecting rods (5). By turning the eccentrics relative to one another, so that their respective throws cancel out, the outer circumference of the pair becomes concentric with the axis of rotation of the cylinder block, and no movement of the pistons in the cylinders occurs when the cylinder block is rotating. If, from this position, the eccentrics are rotated about one another in either direction, the stroke of the pistons will be varied. In the smaller sizes of pumps, another method of stroke variation is adopted, the crank being constructed in one piece and having a slide machined on it, this slide being adapted to move in a rectangular slot in a guide-plate attached to the pump casing.

The drive shaft of the pump is coupled to a cylinder-block extension which forms the shaft and central rotating valve (6). Oil is admitted to and discharged from the cylinders by a fixed valve (7), in which the cylinder-shaft extension rotates, the working fluid being conveyed from

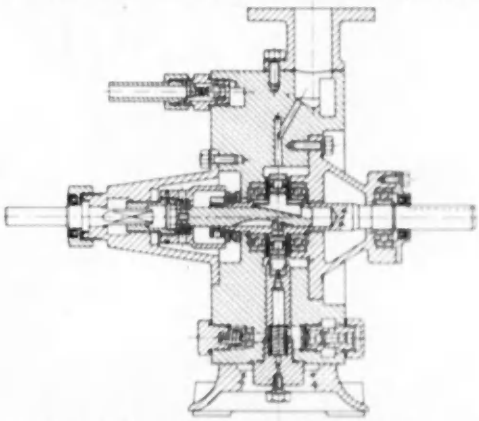
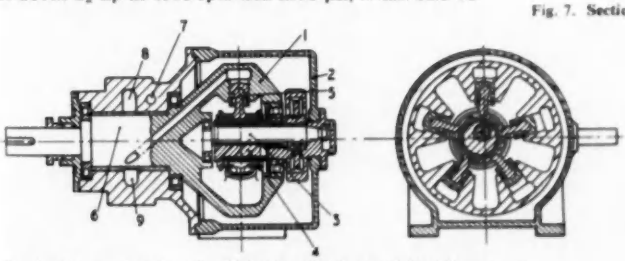


Fig. 7. **Sectional view of a "Mono-Radial" variable-delivery pump.**
(Courtesy of Andrew Fraser & Co. Ltd.)



ig. 6. **Cross-section through a Hele-Shaw Beacham variable-delivery pump.**
(Courtesy of Greenwood & Batley Ltd.)

a suction pipe (8) through passages in the cylinder block to the separate cylinders, whence it is forced back along the passages under pressure to the delivery pipe (9), or *vice versa*, depending on the stroke control position.

The hydraulic motors used are constructed on the same principle, except that the crank is in a fixed position. The direction of rotation of the motor is controlled by the fluid entering by either of

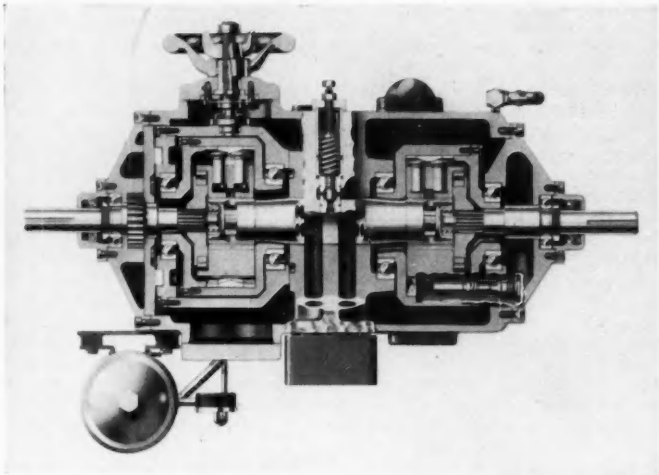


Fig. 8. **Plan-view section of a package-type "Oilgear" variable-speed transmission.** (*Courtesy of The Oilgear Company.*)

the pipes (8) or (9), and exhausting through the other. A wide range of motors for use with these hydraulic pumps is available, with maximum capacities from about 1½ hp at 750 rpm up to about 105 hp at 200 rpm.

Another design using an eccentric on the drive shaft is the "Mono-Radial" variable-delivery hydraulic pump, made in England by Andrew Fraser & Co. Ltd. This unit (Fig. 7) basically consists of a heavy-duty iron cylinder block with pistons radially disposed about a roller bearing mounted on a drive shaft, which is machined and fitted with an infinitely variable eccentric, on which the roller bearing is mounted. Variation in output is obtained by a control at the front end of the pump, operated manually, mechanically, or hydraulically.

"Mono-Radial" variable-delivery pumps are available in a wide range of sizes, i.e., from about 3 to 40 hp, with corresponding continuous-duty capacities of from 1·25 gpm at 940 rpm and 3000 psi to 36 gpm at 940 rpm and 1500 psi. It should be understood, however, that they are intended for use exclusively as pumps, and not as motors, but infinitely variable speed drives can be provided by combining a "Mono-Radial" pump with one of the range of fixed-displacement "Deri Sine" hydraulic motors made by the same company, and described on page 288.

The Oilgear "Any-Speed" variable-speed transmission, made in the U.S.A. by the Oilgear Company, employs bevelled reaction rings, which serve to impart a rolling motion to the radially disposed pistons. In this design (Fig. 8), rotation of the pump unit is transmitted directly to the cylinder barrel, which is mounted on a fixed pintle through a splined floating coupling flange. The radial rolling pistons are confined in the rotor by concave or bevelled reaction rings, while the rotor is carried on anti-friction bearings in the adjustable-stroke slide block.

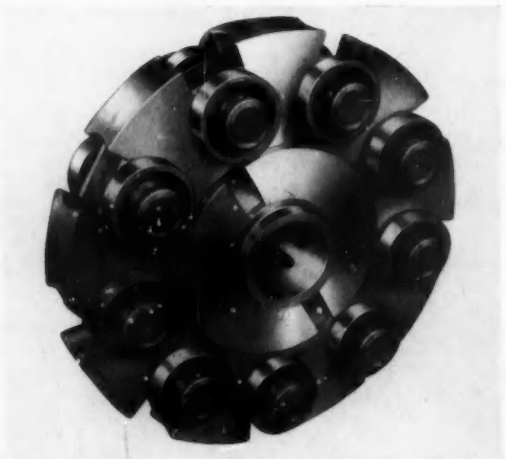


Fig. 9. **Detail of roller assembly of a Pittler hydraulic variable-delivery pump.** (*Courtesy of Hanauer Pumpen-und Getriebebau G.m.b.H.*)

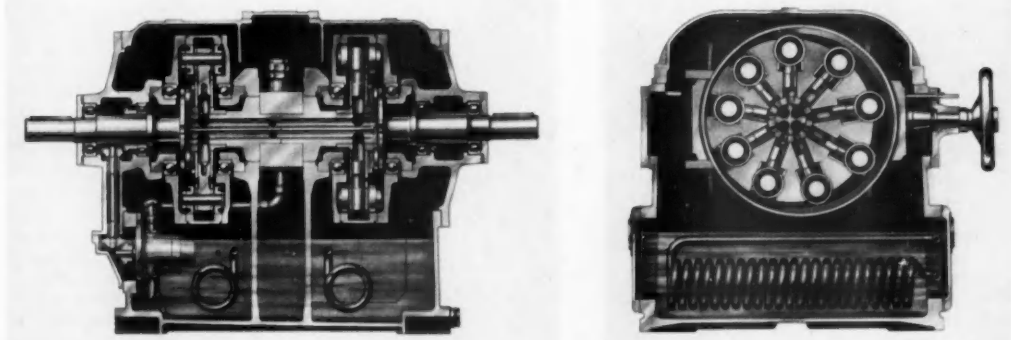


Fig. 10. **Sectional views of a Pittler package-type hydraulic variable-speed drive.** (*Courtesy of Hanauer Pumpen- und Getriebebau G.m.b.H.*)

When the centre-lines of the cylinder and rotor do not coincide, the differences between the radii from the centre of the cylinder to the points of contact of the several piston heads with the bevelled reaction-ring surface cause the piston heads to travel a greater or a lesser distance, depending on their points of contact with the reaction ring. As can be seen from Fig. 8, the use of a bevelled reaction ring ensures that the point of contact of the ring and piston head is offset, so that any difference in distance is taken care of by the slow rotation of each piston in its bore, this rotation being in one direction during half a revolution and in the opposite direction during the other half revolution. Thus, the piston rotates and reciprocates simultaneously. The working parts of the motor unit are identical with those of the pump unit, except that its rotor rotates with a fixed eccentricity.

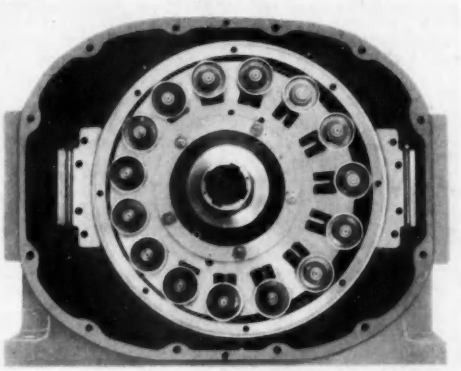


Fig. 11. **Section through a Kieler Howaldtswerke variable-delivery pump, showing disposition of rollers.**
(Courtesy of Kieler Howaldtswerke A.G.)

A built-in differential-type relief valve, adjusted to operate at any pressure within the working capacity of the unit, protects the power input mechanism, the transmission, and the driven machine against overload. Back pressure is maintained at all times by a built-in gear pump, and an oil-filter unit is included.

Oilgear package-type " Any-Speed " units are available in various sizes with normal continuous outputs ranging from about 2½ hp at 1600 rpm up to about 22 hp at 1090 rpm. In addition, separate pump and motor units are available, with maximum capacities up to about 100 hp.

An example of a radial-piston unit using a reaction ring and rollers is the " Pittler ", manufactured in Germany by Hanauer Pumpen-und Getriebebau G.m.b.H. The construction of these units will be clear by reference to Fig. 9, which shows the details of the roller assembly, and Fig. 10, which shows sectional views of a package-type drive comprising combined pump and motor units, although they can also be supplied as separate units. Pittler transmissions are available in a range of sizes with maximum capacities from about 1½ hp at 1500 rpm to about 160 hp at 600 rpm, and include a series of special low-running motors with capacities ranging from about ½ hp to 54 hp at 160 rpm.

As will be appreciated from Figs. 11 and 12, variable-speed units produced in Germany by Kieler Howaldtswerke A.G. also employ rollers and are of virtually identical design to those previously described. These units are available in a very wide range of sizes, with maximum capacities up to about 730 hp at 1500 rpm.

Another radial-piston unit employing rollers but, as can be seen from Fig. 13, with some differences in design, is manufactured in France by S.O.M. The 5-hp model illustrated is a package-type unit, comprising a variable-delivery hydraulic pump and a variable-displacement motor, though this firm also produces a similar unit with a variable-delivery pump and a fixed-displacement motor. In the dual-control unit shown, the pump and motor units are of virtually identical construction, but the motor has approximately twice the displacement of the pump. In operation, the control of the motor unit is set to maximum displacement, and speed is gradually increased from zero up to a maximum of about ±700 rpm by adjusting the pump control to give progressively increasing displacement. For higher speeds, up to a maximum of

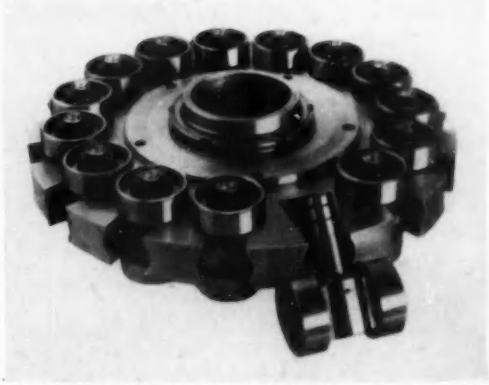


Fig. 12. **Detail of roller assembly of a Kieler Howaldtswerke variable-delivery pump.**
(Courtesy of Kieler Howaldtswerke A.G.)

±3000 rpm, the pump control is left at its maximum-displacement setting, and the motor control is adjusted to give progressively decreasing displacement. Reversal of the drive is effected by operating the pump control in the opposite sense. With this drive, which has a maximum capacity of 5 hp, the output is fully reversible and can be adjusted between zero and twice the input speed.

Other package-type units of higher capacities are also produced by S.O.M., and, in addition, separate pump and motor units (with radial or axial pistons) in a wide range of sizes.

Another radial-piston type of drive is the Carter type

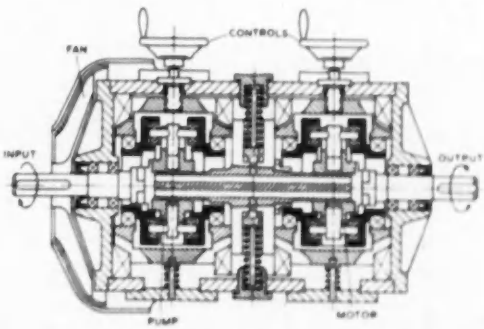


Fig. 13. **Section through a 5-hp package-type S.O.M. hydraulic variable-speed drive.**
(Courtesy of Société d'Optique et de Mécanique de Haute Précision

" A " infinitely variable speed gear, which achieves durability and high efficiency by reducing to a minimum the flow of oil and movement of working parts over the entire speed range. Thus, a feature of the Carter gear is that, at maximum output speed, there is no flow of oil or relative movement of working parts, other than the bearings, all power being transmitted by direct torque from the input to the output shafts.

This action can best be understood by considering a simple piston-type oil-pump unit. If, during the delivery stroke of such a unit, the outlet valve is closed, the resulting oil pressure on the piston and thrust in the connecting rod, caused by the turning moment on the crank, will produce a side thrust on the cylinder block. If the cylinder block is not adequately restrained to resist the resultant turning moment, it will revolve in the same direction and at the same speed, transmitting the same torque as the crank. Similarly, if the turning moment is applied to the cylinder block in the opposite direction of rotation, an equal turning moment will be produced on the crank. This is basically the action of the Carter gear when the input and output speeds are equal, i.e., when the pump unit is at maximum stroke and the motor unit is at zero stroke, so that its pistons cannot accept any oil. This, in effect, closes the outlet valve of the pump unit, so that, as mentioned above, the output shaft of the gear must revolve at virtually the same speed as the input shaft.

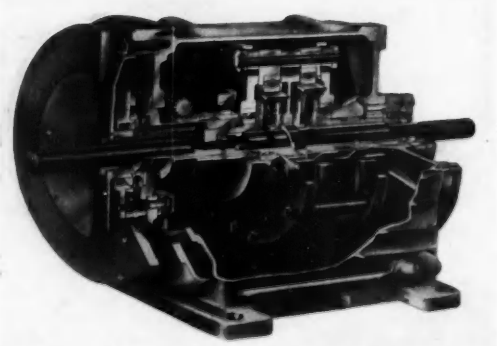


Fig. 14. Cut-away view showing the internal arrangement of the " A " type Carter hydraulic variable-speed gear.
(Courtesy of Carter Gears Ltd.)

Similarly, referring again to a simple piston-type oil-pump unit, if, instead of trapping the oil in the cylinder, it is permitted to pass out at some rate less than the normal capacity of the pump when the cylinder block is stationary, the cylinder block and crank will rotate at different speeds, depending on the amount of oil flow, though each will be capable of exerting a torque which is equal to the applied twisting moment on the other. The oil passing from the cylinder under pressure can be fed to a motor unit to produce a useful turning effort, and by connecting the cranks of both pump and motor units, the turning efforts can be combined. This, then, is the action of the Carter gear at all speed ratios other than 1:1, when all the oil delivered by the pump in a given time is absorbed by the motor in the same time.

With regard to the actual construction of a Carter type " A " gear (Fig. 14), it will be seen that the mechanism comprises two similar multiple-piston units, i.e., the input pump unit and the output motor unit. The pump cylinder block rotates at the constant input speed derived from the input drive motor, and the motor cylinder block is rigidly fixed to the casing of the gear. Eccentric or crank elements, in the form of ball-bearing races for controlling the strokes of the pistons of both units, are set at 180 deg. and are movable across the axis of the gear simultaneously, so that the sum of the strokes is always constant. These eccentric elements are connected to and rotate at the same speed as the variable-speed output shaft, which also forms the oil distributing valve, thus ensuring correct phasing of the oil supply with the piston strokes.

Movement of the eccentric-element unit for achieving speed control is effected as follows :—A control shaft projecting from the casing of the gear, at right angles to the axis of the input and output shafts, carries at its outer end a lever, or a quadrant of a worm-wheel, forming the connection for manual, remote, or automatic control. At its inner end, the control shaft is in the form of a spur pinion engaging with a rack formed in a sleeve, which is movable in the line of axis of the driving shafts. This sleeve transmits its movement by means of a ball-bearing race to a double-armed casting, which revolves with the casing carrying the eccentric ball-bearing elements controlling the stroke of the pump and motor pistons. The outer ends of the arms of this casting are each connected to a bar sliding axially in the revolving casing. In each of these bars is cut diagonally a number of teeth which engage with similar teeth cut in components attached to the housings of the eccentric bearing. The interaction of the two pairs of racks translates the axial motion into the crosswise motion controlling the positions of the two eccentric-bearing elements. The whole assembly is housed in a compact fan-cooled dust-proof casing with no external pipe connections or joints under working pressure, and the internal mechanism, although not immersed in or dipping into the oil sump, is entirely self-lubricating.

It should be noted that, under load conditions, the output speed of any hydraulic gear is slightly lower than that indicated by the relative strokes of the pump and motor units, this being due to the loss of oil under pressure past the piston and valve-shaft clearances. In the Carter " A " gear, a low-pressure make-up pump is incorporated at the outer end of the gear to offset this loss of oil and also to provide pressure lubrication for the input bearing. The main hydraulic system is fitted with non-return valves, so that the gear is suitable for either direction of rotation and for use in reversing drives without modification, while the make-up pump is also designed to reverse its porting with reversal of the drive shaft.

Carter " A " gears are made in a variety of sizes, with capacities ranging from about 1·7 hp at 1350 rpm to about 21½ hp at 940 rpm. Although all these gears will pick up load from zero rpm, there is a certain recommended minimum running speed in each case, the ratio of maximum to minimum speed being approximately 27:1. These units can also be supplied with flange-mounted or direct-coupling reduction gears, or as complete drives.

Another radial-piston design of hydraulic variable-speed gear made by Carter is their " F "-type, which is basically intended for low-power applications. In many respects, the design of the " F "-type gear is similar to that of the " A " type, but, whereas the latter has a pump unit which can be operated in either direction of rotation, reversal of the motor unit being accomplished by reversal of the pump unit, the pump unit of the " F "-type gear is intended for rotation in one direction only (i.e., either

clockwise or anticlockwise, depending on requirements), reversal of the output shaft being effected by reversal of the motor unit. Thus, the " F "-type gear provides infinitely variable speed regulation from maximum in one direction through zero to maximum in the opposite direction. The gear is capable of picking up a load from zero speed and of providing smooth variable output speeds from about 35 to 50 rpm minimum, depending on the model.

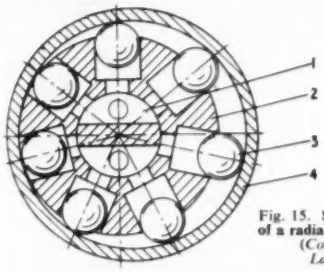


Fig. 15. **Schematic arrangement of a radial-type ball-piston pump.** (*Courtesy of La Précision Mécanique.*)

Like the " A "-type gear, the Carter " F "-type gear is housed in a compact casing, and incorporates a small make-up pump. In the " F "-type gear, this pump, in addition to replacing oil lost under working pressure past the pistons and valve-shaft clearances, also serves to ensure lubrication of the valve shaft when the gear is running at zero output speed, to provide a source of hydraulic power for a servo-control unit, and to eliminate the use of springs for keeping the pistons in contact with the thrust rings. Various sizes of " F "-type gear are available, with capacities ranging from about $\frac{1}{8}$ hp at 1420 rpm to about 3 hp at 960 rpm.

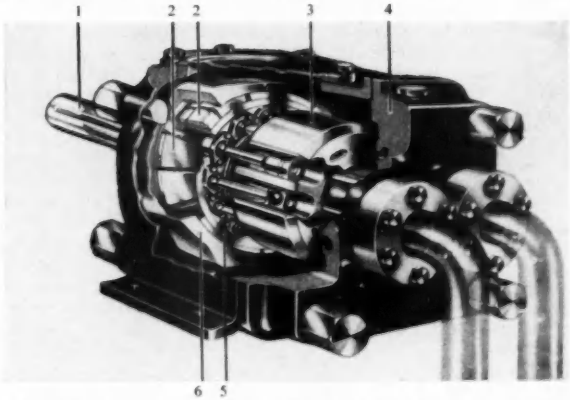


Fig. 16. **Cut-away view of a typical " VSG " hydraulic variable-delivery pump.** (*Courtesy of Vickers-Armstrongs Ltd.*)

An interesting design of radial-piston pump, produced in France by La Précision Mécanique, is shown schematically in Fig. 15. In this design, the cylinder block (2) rotates about the pintle (1) in the conventional manner, but these are made of balls (3) which serve the dual purpose of acting as pistons and of rolling along the adjustable reaction ring (4), with which they are held in contact by centrifugal force. These pumps (Keller patent) are supplied combined with a motor of identical design in the form of a fully reversible package-type unit, the maximum capacity for continuous operation being about 4 hp at ±2000 rpm. A similar unit, with the addition of one of a range of reduction gears, is also available. An inherent feature of this ball-piston design is the use of low pressures (of the order of about 375 psi); as a result, construction is comparatively economical, as the degree of manufacturing accuracy necessary with high-pressure units is not in this case required. Incidentally, it has been reported by the Mechanical Engineering Research Laboratories at East Kilbride that specially designed transmissions using the ball-piston principle have been operated at speeds up to 24,000 rpm, though only for small powers.

Axial-Piston Types

In pumps and motors using axial pistons, the cylinders are arranged in a rotating barrel with their axes parallel or slightly inclined to the axis of rotation, the basic idea being to ensure compactness of the assembly and to keep the outside diameter of the barrels as small as possible for high-speed applications. In this arrangement, an inclined rotating swashplate is used for operating the pistons via individual connecting rods, variable piston displacement being obtained by tilting the swashplate about its axis. Alternatively, the swashplate can be replaced by a fixed flange rigidly connected to the drive shaft and the rotating cylinder body swung with reference to the flange, this being essentially the design developed by Professor Thoma.

In the " VSG " hydraulic pumps and motors produced in England by Vickers-Armstrongs Ltd., the tilting-swashplate method is used. The cylinders in the pump are formed, as shown in Fig. 16, in the cylinder barrel (3), which is held in contact with the valve plate (4) by hydraulic pressure, initial contact being maintained by a spring. Each piston is connected to a socket ring (5) by a ball-ended rod, whose ends are held in sockets formed by the piston and the socket ring respectively. The cylinder block and the socket ring are both con-

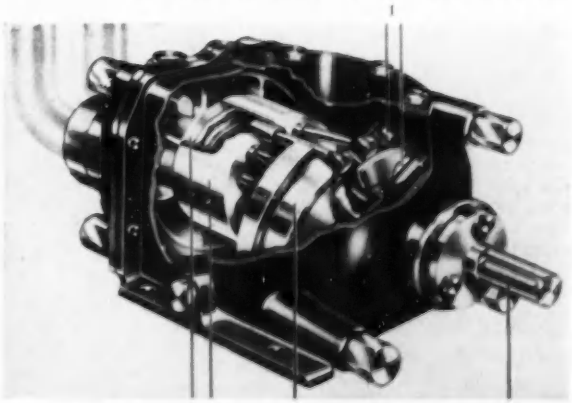


Fig. 17. **Cut-away view of a typical " VSG " hydraulic motor.** (*Courtesy of Vickers-Armstrongs Ltd.*)

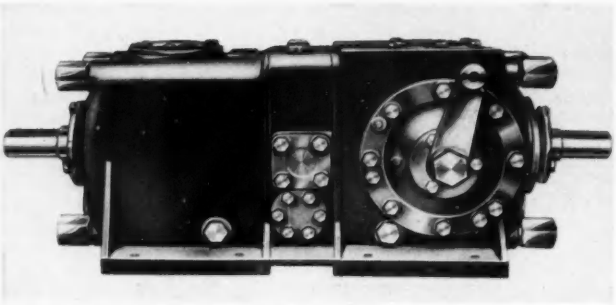


Fig. 18. Combined " VSG " hydraulic infinitely variable transmission unit with a common valve plate. *(Courtesy of Vickers-Armstrongs Ltd.)*

nected to the drive shaft (1) by separate universal joints, so that, as the shaft rotates, the cylinder block and socket ring both rotate with it. The socket ring rotates inside the swash-plate assembly or tilt-box (6), the bearing between the two parts being provided by two sets of Michell pads (2), which take the axial and radial loads resulting from the oil pressure on the ends of the pistons.

It will be appreciated that the quantity of oil delivered per revolution of the pump depends on the angle of tilt of the swashplate. Thus, if the swashplate is vertical, no oil is delivered, as there is no axial movement of the pistons. If the swashplate is tilted in one direction, there is an immediate flow of oil. If the swashplate is tilted in the opposite direction, the direction of oil flow is reversed, even though the direction of rotation of the drive shaft (1) remains unchanged.

The hydraulic motor of the " VSG " drive (Fig. 17) can be considered as a replica of the pump unit, with the exception that roller bearings (1) are used in place of the Michell pads. Here, of course, the oil delivered under pressure from the pump acts via the valve plate (5) against the pistons in the cylinder barrel (4), and the output shaft (2) is rotated by the force transmitted by the connecting rods to the socket ring. The " VSG " hydraulic motor unit shown in Fig. 17 is of the constant-capacity type, in which the swash-plate assembly (3) is held at a pre-determined fixed angle, so that the volume of oil used to rotate the motor unit cannot be varied. For some applications, however, variable-capacity motor units can be supplied, in which the swashplate can be rotated about its axis in the same way as in the pump unit. In such cases, where variable displacement is provided for both the pump and motor units, it will be understood that the output speed of the motor unit is dependent on a combination of the tilt settings of both swashplates. The tilt of the pump swashplate determines the amount of oil flowing in the system but, since an alteration of the setting of the swashplate in the motor unit changes the capacity of the motor, i.e., the quantity of oil required to turn it through one revolution, a decrease in the tilt of the motor-unit swashplate increases the speed of the motor for a given tilt of the pump swashplate.

At the same time, for a given oil pressure, the torque at the output shaft of the motor varies with motor-cylinder capacity. A decrease in the tilt of the motor swash-plate will therefore result in an increase in motor speed and a decrease in torque, thus giving the same out-put horsepower, provided, of course, that the setting of the pump swashplate and the oil pressure remain con-stant. It will thus be seen that, if the tilt of the motor swashplate is controlled in such a way that the oil pres-sure remains constant, a constant-horsepower output will be obtained over a considerable speed range. This feature is of advantage in several applications, e.g., in coiler drives, where constant strip tension and linear strip speed are required, irrespective of coil build-up.

These " VSG " units are available in various sizes, with maximum outputs over the upper half of the speed range of from about 2½ hp at 960 rpm to 150 hp at 400 rpm, and corresponding maximum starting torques from about 42 to 6000 lb-ft. In general, the pump and motor are separate units, but they can, in certain circumstances, be combined to form a single unit with a common valve plate, as shown in Fig. 18. In all cases, an infinitely

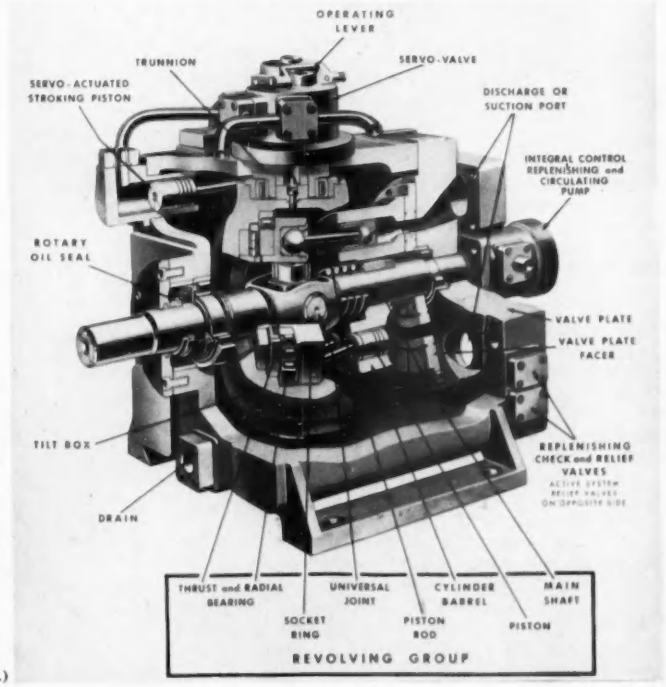


Fig. 19. Cut-away view of a Waterbury variable-delivery heavy-duty axial-piston pump. *(Courtesy of Waterbury Tool Division of Vickers Inc.)*

variable speed range from zero to maximum in either direction is possible by operation of an external lever or other suitable control connected to the tilt-box.

Another example of a positive-displacement, axial-piston design with a movable swashplate is the Waterbury heavy-duty variable-displacement pump shown in Fig. 19, which illustrates clearly its basic constructional features. Here again, the rate and direction of flow depend on the angle of the tilt-box. For variable-speed applications, these pump units, which are made in the U.S.A. by the Waterbury Tool Division of Vickers Inc., are intended for use with corresponding Waterbury hydraulic motors, these being of similar design to the pump units. As in the case of the " VSG " units previously described, these motor units are available as fixed- or variable-displacement types, the most common combination being a variable-delivery pump and a fixed-displacement motor for constant torque characteristics. Similarly, a combination of constant-delivery pump and variable-displacement motor can be used for constant horsepower characteristics, or a variable-delivery pump and variable-displacement motor for any desired variation of torque and horsepower over a very wide speed range, the upper speed range being at constant horsepower and the lower range at constant torque. Particularly adapted to heavy-machinery drives requiring precise control of torque, acceleration, deceleration, speed, or reversal, these units are available with output capacities

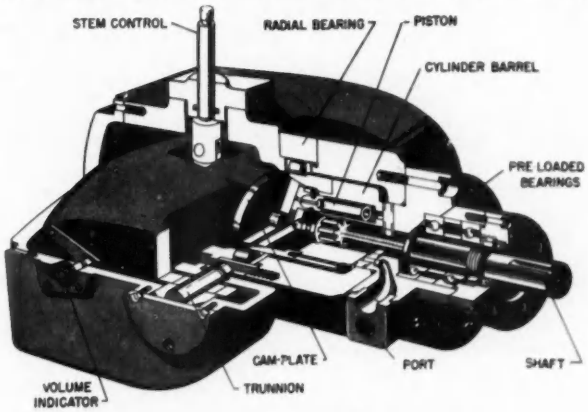


Fig. 20. **Cut-away view of a Denison " Hydroilic " variable-delivery axial-piston pump with stem control.**
(Courtesy of Denison Engineering Division, American Brake Shoe Company)

ranging from about 45 hp at 1200 rpm up to about 3600 hp at 400 rpm.

A very wide range of variable-delivery hydraulic pumps is made in the U.S.A. by the Denison Engineering Division of the American Brake Shoe Company. These units are also of the axial-piston type employing an adjustable cam-plate or swashplate, and are available with various forms of control, including stem control (Fig. 20) and handwheel control (Fig. 21). The combination of one of these pumps with a separate fixed-displacement motor of similar design provides a reversible and infinitely variable speed transmission. These " Hydroilic " motors are available in a range of sizes with maximum capacities from about 1·8 hp per 100 rpm at maximum pressure (5000 psi) to 9 hp per 100 rpm at maximum pressure, corresponding to between 55 and 180 hp at maximum pressure and speeds (3000 and 2000 rpm respectively).

Another design of axial-piston pump with an adjustable swashplate is that produced in the U.K. by Joseph Lucas. Stemming from their original design of variable-

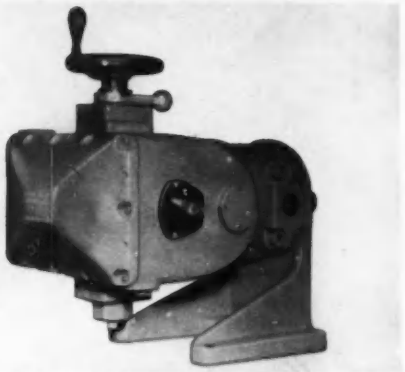


Fig. 21. **Denison " Hydroilic " variable-delivery axial-piston pump with handwheel control.**
(Courtesy of Denison Engineering Division, American Brake Shoe Company)

delivery hydraulic pump extensively used in aircraft gas-turbine fuel systems, these industrial units are at present available in three models, i.e., IP.60, IP.525, and IP.3000. Of these, for variable-speed drives, only type IP.60 is at the moment intended for use with its corresponding type IP.60 hydraulic motor (Fig. 22), which is of identical design to the pump unit, but has a fixed swashplate. The capacity of this motor for continuous operation is about 3½ hp at 3000 rpm. The corresponding motors for the other two pump units are not as yet available, but are under development. Incidentally, the type IP.525 pump is rated at 525 gph per 1000 rpm, with a maximum working pressure and speed for continuous duties of 2000 psi and 2000 rpm respectively, while the type IP.3000 pump is rated at 3000 gph per 1000 rpm, with a maximum working pressure and speed for continuous duties of 2000 psi and 1500 rpm respectively. A cut-away view of the type IP.3000 variable-delivery pump is shown in Fig. 23. It should be noted that, although the range of variable-

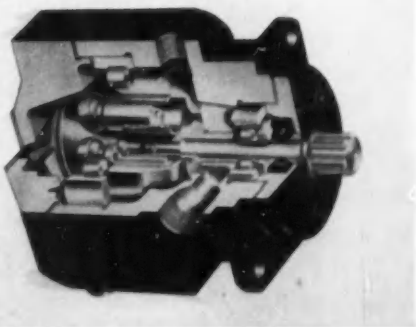


Fig. 22. **Cut-away view of a Lucas IP.60 hydraulic motor.**
(Courtesy of Joseph Lucas (Hydraulic & Combustion Equipment) Ltd.

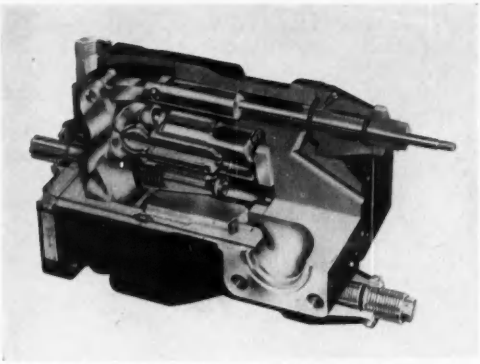


Fig. 23. **Cut-away view of a Lucas IP.3000 variable-delivery pump.** (*Courtesy of Joseph Lucas (Hydraulic & Combustion Equipment) Ltd.*)

speed hydraulic equipment made by Joseph Lucas is not at present extensive, this firm is rapidly expanding its operations in this field and is devoting considerable research not only into the development of industrial hydraulic variable-speed drives, but also into hydrostatic transmissions for the automobile industry.

As distinct from many other variable-displacement pumps and motors, which are so designed as normally to assume a position of maximum displacement, the control forces being applied to move the pump and motor to a position of minimum displacement, the Sundstrand type 32PV and 32MV variable-displacement axial-piston pumps and motors, produced in the U.S.A. by the Sundstrand Machine Tool Co., embody the opposite principle, i.e., they normally assume a position of minimum displacement, and the control forces are applied to move the swashplates of the pump and the motor to the position of maximum displacement. This makes control problems more difficult, but has a number of advantages. Thus, control response is rapid, owing to the opposing forces set up in the controls, and " fail-safe " control is provided, as the pump swashplate can be moved to the neutral or minimum position merely by releasing pressure on a power piston, thus unloading the system. In addition, the control pressure is approximately 50% of the working pressure. Two basic types of control are used in these Sundstrand variable-delivery units, i.e., a flow-sensitive control and a pressure-sensitive control. The flow-sensitive control uses a variable orifice (usually

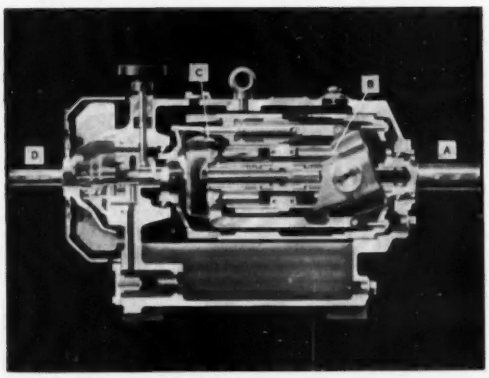


Fig. 24. **Cut-away view of the " Varicon " package-type hydraulic infinitely variable speed drive.** (*Courtesy of David Brown Corporation (Sales) Ltd.*)

remote from the pump) to control the displacement of the pump, while the pressure-sensitive control serves to put the motor into the position of maximum displacement at a given pressure signal. The maximum capacity of these units is about 18 hp at 1400 rpm.

The " Varicon " infinitely variable hydraulic drive, made under licence in the U.K. by David Brown, is shown in Fig. 24. Based on the original Italian design by Badalini, this package-type unit incorporates a hydraulic pump *B* and a hydraulic motor *C*, each of the positive-displacement, axial-piston type, located in line, with their cylinder heads separated by a central distributor plate which transfers oil from the pump to the motor and also takes the reaction of the oil pressure on the pistons. Displacement of the pistons is controlled by swashplates, which are adjusted through linkage by a control handwheel, the angular settings of the swashplates being actuated simultaneously by a servo-cylinder and a coupling rod, both of which are carried in the frame of the output shaft *A*. The oil required to operate the servo-piston is supplied by the main pressure pump.

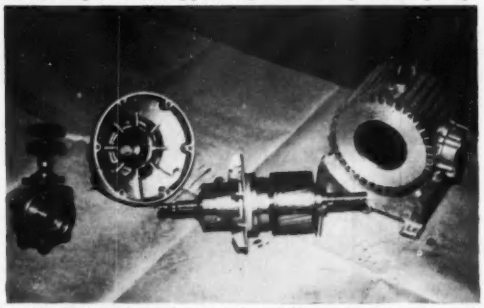


Fig. 25. **Badalini hydraulic infinitely variable speed unit incorporated in a standard electric motor.** (*Courtesy of Flender-Cigala*)

When the swashplates are set to give maximum stroke to the pump pistons and zero stroke to the motor pistons, a hydraulic lock exists, as the motor pistons are unable to reciprocate and oil cannot then flow from the pump to the motor, this principle being similar to that embodied in the Carter radial-piston design already described. The torque transmitted to the pump cylinder body is thus passed directly to the inclined swashplate housing, the reaction of which rotates the output shaft *A* at a speed closely approximating that of the input speed. In fact, a small loss in speed occurs, not exceeding 1% of the input speed, and the drive with the unit set to this ratio is wholly mechanical.

Oil is supplied by a primary pump located on the input shaft *D* and is circulated from the sump through the shaft sleeve to the cylinder bodies, the supply being drawn through a magnetic oil filter. A fan is fitted to the input shaft and projects cool air along horizontal ribs cast on the case.

At present, the " Varicon " is available in one model with a maximum output of about 5 hp at 1500 rpm, but a 10-hp model will shortly be available, and possibly also other models later on.

The Badalini drive is also manufactured under licence in Italy by Flender-Cigala under the name of " Idrovar " and in Germany by Flender-Bocholt under the name of " Hyvari ". Basically, these drives are similar, though certain minor modifications have been incorporated. In the smaller sizes of " Idrovar " and " Hyvari " units (up to 10 hp inclusive), the pistons are parallel, whereas

in the larger sizes they are inclined, partly because, in the inclined arrangement, centrifugal force assists the return stroke of the pistons, and partly because this design conserves space. Various sizes of " Idrovar " and " Hyvari " drives are available, with capacities ranging from about 2½ to 50 hp at 1500 rpm for the " Idrovar ", and from about 4 to 30 hp at 1500 rpm for the " Hyvari ".

An outstanding feature of the Badalini unit is its compactness. Thus, in addition to an extremely compact infinitely variable hydraulic transmission unit incorporated in an Italian MV Agusta motor-cycle engine, another unit has been specially designed as an integral part of a standard electric motor (Fig. 25), delivering a maximum of about 1½ hp, with speeds infinitely variable between zero and 1800 rpm.

An example of an axial-piston type of hydraulic variable-speed drive in which the cylinder body is swung with reference to a fixed flange is the " Hydrogigant " drive, which is made in Germany by Hydromatik G.m.b.H. In this unit, the pump (Figs. 26, 27, and 28) comprises an outer casing bolted to the pump body (8), and free to swivel around a hemispherical seating carried by the drive shaft (1). The shaft is fitted with a flange in which are located the spherical heads of the connecting rods (2). The rods (2) and the pistons (3) connected to them also rotate with the shaft (1), thus causing the cylinder block (4) to rotate as well. As a rule, seven axially parallel cylinders are used, and the cylinder block (4) is guided by a central pin (5), its spherically shaped end face being supported by a similarly shaped control cone (6), which is screwed to the pump body (8) by means of a ring nut (7), as shown in Fig. 26. The pump body and control cone do not rotate.

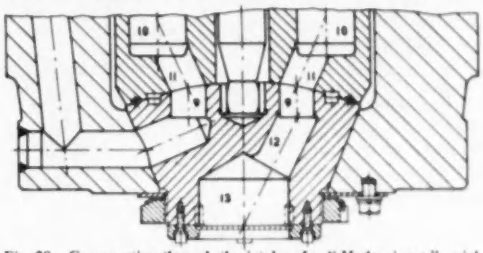


Fig. 28. **Cross-section through the intake of a " Hydrogigant " axial-piston pump with infinitely variable delivery.**

(9) Kidney-shaped recess. (11), (12) Channels.
(10) Cylinder bore. (13) Suction port.

(Courtesy of Hydromatik G.m.b.H.)

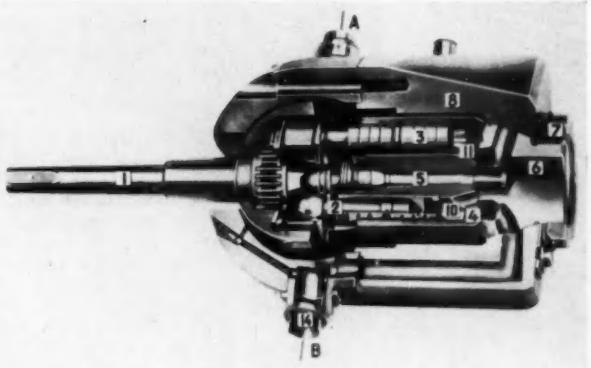


Fig. 26. **Cut-away view of a " Hydrogigant " axial-piston pump with infinitely variable delivery.**

(1) Drive shaft. (5) Central pin. (10) Cylinder bore.
(2) Piston rod. (6) Control cone. (11) Channel.
(3) Piston. (7) Ring nut. (14) Sealed bushing.
(4) Cylinder block. (8) Pump body. *AB* Swivel axis.

(Courtesy of Hydromatik G.m.b.H.)

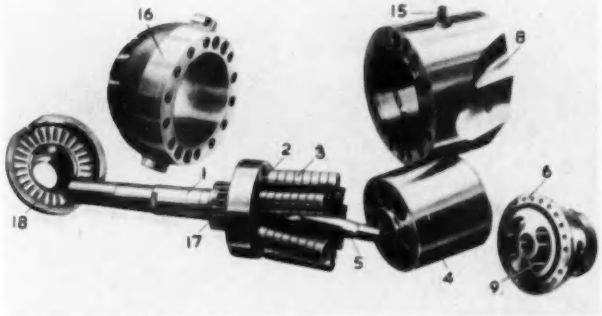


Fig. 27. **Exploded view of a " Hydrogigant " axial-piston pump with infinitely variable delivery.**

(1) Drive shaft. (5) Central pin. (15) Pivot.
(2) Piston rod. (6) Control cone. (16) Spherical cap.
(3) Piston. (8) Pump body. (17) Radial bearing.
(4) Cylinder block. (9) Kidney-shaped recess. (18) Axial bearing.

(Courtesy of Hydromatik G.m.b.H.)

When the cylinder block is parallel to the drive shaft (1), the stroke of the piston is zero. When, however, it is swivelled to one side around the axis *AB*, the stroke of the piston is increased steadily. When swivelled to the other side, the direction of delivery is reversed, i.e., the pressure and suction sides are interchanged.

The control cone (6), as shown in Fig. 27, is fitted with two kidney-shaped recesses (9), and the cylinder bores (10) in the cylinder block open out on the side of the control surface into channels (11) which are directed at an angle towards the inside. These channels slide over the kidney-shaped recesses (9) when the cylinder block rotates. During half a revolution, oil is sucked into the cylinders through one of the two recesses ; during the second half of the revolution, the oil is discharged through the other recess. Normally, the recess on the suction side of the control cone (6) is linked directly with the suction port (13) by a channel (12), as shown in Fig. 28, thereby facilitating free flow. When operation is at high speeds, it is often necessary to dispense with suction at atmospheric pressure, and in this case the oil is introduced via a sealed bushing (14) and the suction port (13) is not used. Oil feed is then effected by means of a gear pump which boosts the entire pump system to a pressure of a few atmospheres.

In principle, the hydraulic motor of the " Hydrogigant " unit is of the same design as that of the pump, but no provision is usually made for adjustment of piston stroke, though units with a variable piston stroke are also available. Similarly, they can be supplied as

separate pump and motor units, or as combined units. Maximum continuous capacities of " Hydrogigant " variable-speed drives range from about 3 hp at 1500 rpm to about 120 hp at 750 rpm, with approximately 50% increase in output for intermittent operation.

The " Hydrogigant " drives just described are based on the original design of Professor Thoma. Also of Thoma design, and basically similar to the " Hydrogigant " drives are the high-pressure axial-piston units manufactured in Germany by Mannesmann-Meer. However, these units are intended for high-power applications, ranging from about 100 hp at 1200 rpm to about 1000 hp at 350 rpm. A typical Mannesmann-Meer pump unit is shown in Fig. 29. Successful applications of these drives include their use in heavy machine tools, in rolling mills, in mine hoists, in cranes, in locomotives, and in marine propulsion.

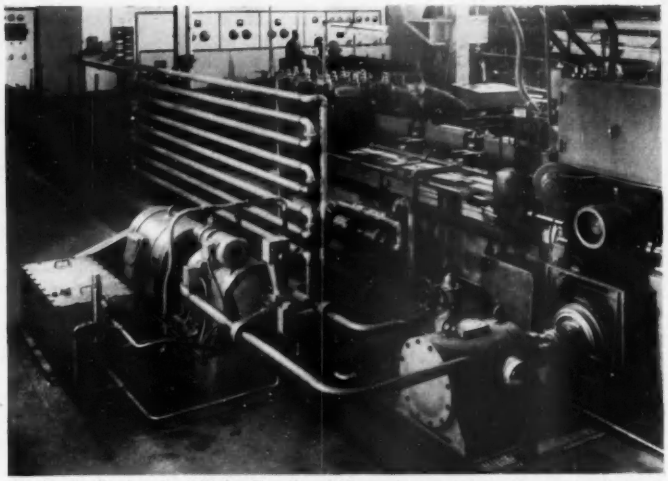


Fig. 31. **Application of a " Hydro-Titan " variable-speed transmission, with separate pump and motor units, to a tube-welding machine.**
(Courtesy of Gesellschaft der Ludw. von Roll'schen Eisenwerke A.G.)

In Switzerland, the Thoma design is represented by the " Hydro-Titan ", manufactured by Gesellschaft der Ludw. von Roll'schen Eisenwerke A.G. (Louis de Roll Ironworks Ltd.). Available in a wide range of separate pump and motor units, with maximum capacities up to about 200 hp, they are also available as combined infinitely variable speed packaged units (Fig. 30), with provision for adjusting the displacement of either the pump or the motor unit, or of both units, so as to obtain constant torque in the lower part of the speed range and constant horsepower in the upper part. These units are normally available in various sizes, with capacities for continuous operation from about 1½ hp at 4000 rpm up to about 42 hp at 1800 rpm, with an increase in capacity of about 50% for intermittent operation. These drives have been used successfully in a wide range of applications, including wire and cable manufacture, conveyors, mining and smelting operations, the petroleum industry, paper and textile manufacture, and metalworking. A " Hydro-Titan " variable-speed transmission, with separate pump and motor units, for a tube-welding machine, is shown in Fig. 31. In this drive, which develops 70 hp, the pump unit is remotely controlled via a servo-mechanism, and the motor unit is hand-controlled. In addition, special " Hydro-Titan " drives have been produced and have been successfully applied in diesel locomotives up to about 350 hp.

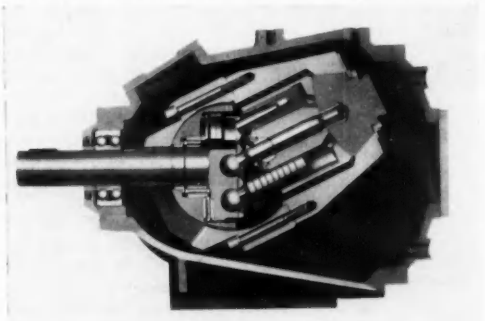


Fig. 29. **Cross-section through a Mannesmann-Meer axial-piston variable-delivery pump.**
(Courtesy of Mannesmann-Meer A.G.)

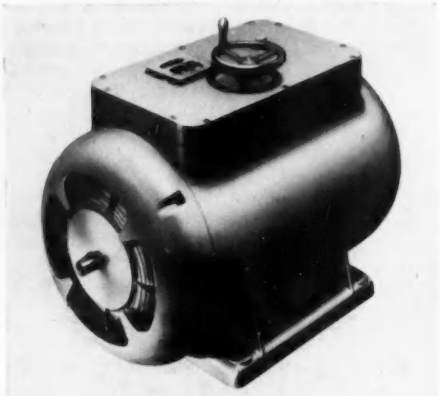


Fig. 30. **View of typical " Hydro-Titan " infinitely variable speed package-type unit.**
(Courtesy of Gesellschaft der Ludw. von Roll'schen Eisenwerke A.G.)

The " Hydro-Stabil " drive, produced in Germany by

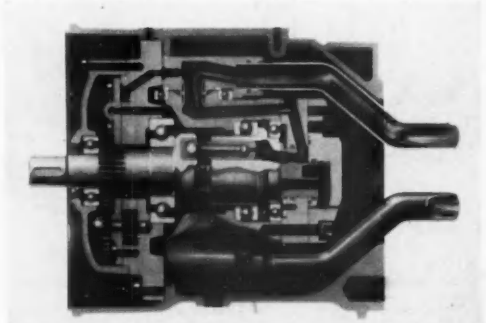


Fig. 32. **Cross-section through a typical " Hydro-Stabil " axial-piston variable-delivery pump.**
(Courtesy of Gusswerk Paul Saalmann & Sohn G.m.b.H.)

Gusswerk Paul Saalmann & Sohn G.m.b.H., is also of basic Thoma design, employing a fixed flange and a tilting housing (Fig. 32). In this design, however, the control cone ((6) in Fig. 27) is replaced by a hydraulic piston, permitting automatic compensation of pressure on the control face, which in this case is flat. It is claimed that this design permits a more favourable dimensioning of the control elements, owing to a reduction in the forces required for swivelling the cylinder body. These drives,

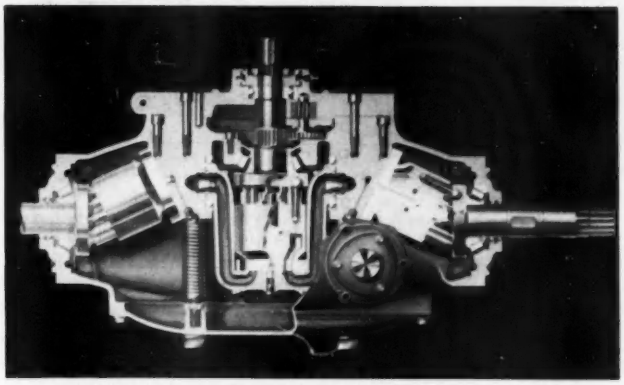


Fig. 33. Combined automotive transmission unit with one pump and two motors.
(*Courtesy of Gusswerk Paul Saalmann & Sohn G.m.b.H.*)

which are available with normal ratings from about 3 hp at 2000 rpm to about 40 hp at 1450 rpm, have been particularly successful in vehicle drives, e.g., industrial trucks. In applications of this type, it will be appreciated that braking can be effected merely by locating the pump body in the neutral position, when there is no further delivery of oil, thus blocking the motors. It is also interesting to note that a special " Hydro-Stabil " unit has been produced, as shown in Fig. 20, in which one pump is used for driving two motors, the whole unit being mounted in the same location and replacing a conventional mechanical differential.

An example of a hydrostatic transmission specially developed for industrial and agricultural vehicles is shown in Fig. 34. In this unit, which is made in the U.K. by Dowty Hydraulic Units Ltd., a ten-cylinder variable-delivery pump, also of basic Thoma design, i.e., with a flange and tilting body, drives a fixed-swash-angle motor, the pump being driven by the normal prime mover, while the motor drives through a reduction gear (or differential). Alternatively, one pump can be used to drive two motors, one on each drive wheel. A gear pump, also driven by the prime mover, supplies and maintains a base pressure of approximately 75 psi in the system, and this pressure resets the main pump to zero swash-angle when the engine is started. In addition, the gear pump operates a low-pressure booster circuit which prevents aeration in the system and assists in circulating cool oil. Direction of movement and speed variation are obtained by a single lever which actuates a servo-control, providing a high degree of manoeuvrability. In order to off-load the system when starting or idling, provision is made for a release control valve, which can be operated by a lever mechanically linked to the speed-control lever in such a way that the speed control is set to zero before the release control can be operated and is locked when the release control is operated. When the release control valve is functioning, it causes the high-pressure oil to by-pass the motor and permits it free passage back to the reservoir. A relief valve is also incorporated, to ensure that oil is directed to the reservoir when the pressure exceeds 4000 psi.

Another design of variable-delivery axial-piston pump in which the cylinder body is swung with reference to a fixed flange is made in the U.S.A. by Vickers Inc., and in the U.K. under licence by the Sperry Gyroscope Co. Ltd. Basically designed for aircraft applications and intended for use with motors of similar design, but of fixed displacement, these pump units are essentially compact and of specially light-weight construction. Two pump housings are available, i.e., permitting movement of the yoke to 30 deg. maximum off centre, and to both sides of the centre, respectively. This latter type allows oil to be delivered under pressure from either of the two pump ports, without the need for special valving arrangements, and provides a fully reversible feature. These variable-delivery pump units are available in several sizes, with maximum capacities at 2700 psi of from about 3 gpm at 7300 rpm for continuous operation or 3¾ gpm at 9100 rpm for intermittent operation, up to about 40 gpm at 2500 rpm for continuous operation, or 50 gpm at 3150 rpm for intermittent operation. The corresponding fixed-displacement motors are available with capacities for continuous operation ranging from about 2½ hp at 9100 rpm up to about 60 hp at 3150 rpm.

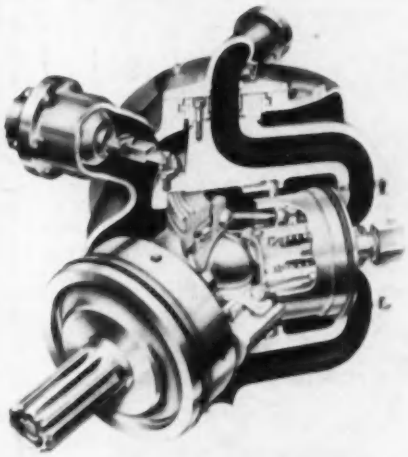


Fig. 34. Cut-away view of the Dowty axial-piston variable-delivery pump for industrial and agricultural vehicles.
(*Courtesy of Dowty Hydraulic Units Ltd.*)

With regard to axial-piston units in general, it may be said that considerable research has been and still is being devoted to improving their design and performance. To-day, axial-piston units have a very high efficiency, and improved design has made it possible to utilize a comparatively small number of cylinders, e.g., seven, resulting in a more compact design of cylinder block and thereby making these units even more suitable for high-speed applications.

Vane Types

In its basic form, the vane-type pump or motor comprises a housing in which, as shown in Fig. 35, a rotor, eccentrically mounted with respect to the housing and fitted with slots carrying sliding vanes, is fitted. The space between the rotor and the housing progressively varies, increasing as the vanes proceed from the point of closest contact between the housing and rotor, and then decreasing after the point of maximum distance between the housing and rotor has been passed. The vanes slide on a track inside the housing, against which they are retained by centrifugal force, and oil is alternately drawn into and discharged from the pump space by opposed valve ports.

In pumps of this design, unbalanced loads of considerable magnitude can easily arise; as a result, balanced-type vane pumps have also been designed, in which two intakes and two outlets, diametrically opposed to each other, are provided, ensuring that the hydraulic loads on the rotor are balanced. Such a design is the Vickers vane pump, made in the U.S.A. by Vickers Inc., and in the U.K. under licence by Stein Atkinson Vickers Hydraulics Ltd. Although this pump is exclusively of the fixed-displacement type and, as such, is not, strictly speaking, in the scope of this survey, it may be regarded as the classic example of balanced vane-pump design, and for this reason is briefly described here.

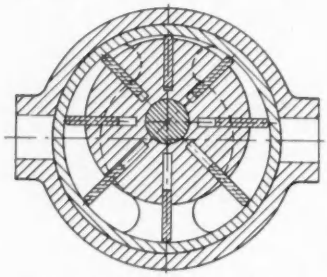


Fig. 35. Schematic arrangement of a typical unbalanced vane-type pump.

As shown in Fig. 36, the Vickers pump utilizes slanted vane slots, permitting the use of longer and better-supported vanes, and is therefore non-reversible, though it is easy to adapt it for either direction of rotation merely by appropriate assembly of the rotor cartridge, which consists of the rotor, the vanes, the valve-plate bushings, and the cam ring. All moving parts, with the exception of the shaft and its bearings, operate within this cartridge assembly, so that no moving parts are in contact with the body of the pump. The pressure chamber is elliptoid, so that two opposing pumping chambers are formed, with side valve-plate bushings for intake and discharge oil.

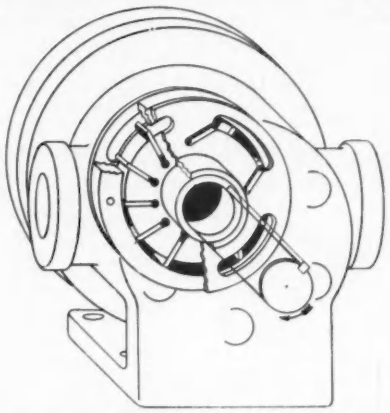


Fig. 36. Cut-away view from the shaft end of a Vickers fixed-displacement vane-type pump.
(Courtesy of Stein Atkinson Vickers Hydraulics Ltd.)

As already stated, these pumps, which are available in a wide variety of sizes and types, including double pumps and two-stage pumps, are of fixed displacement and are normally intended for use with a fixed-displacement motor and suitable valving, i.e, a relief valve, a flow control valve, and a directional control valve, if required. For this purpose, two basic sizes of Vickers vane-type fixed-displacement hydraulic motors, of similar design to the pump units but with radially disposed vanes, instead of slanted vanes, so as to permit reversibility, are available, with capacities for continuous operation from 5 hp at 2200 rpm up to 2 hp at 1800 rpm.

An example of a vane-type variable-speed hydraulic transmission is shown in section in Fig. 37, in which the input shaft is at the left of the illustration. Basically, this design, which is manufactured in the U.K. by Churchill-Sturm Limited, under licence from the German manufacturers, Gebr. Boehringer G.m.b.H., is a package-type unit in which the oil pump and the motor are enclosed in a common casing, which also serves as a reservoir for the oil. Both the pump and motor consist of a rotor fitted with sliding vanes and rotating inside a housing in a frame which is adjustable transversely inside the casing, so that the eccentricity of the housing relative to the rotor can be varied, as required. The ball-bearing-mounted rotors revolve around a fixed central shaft in which are formed channels for oil delivery and return.

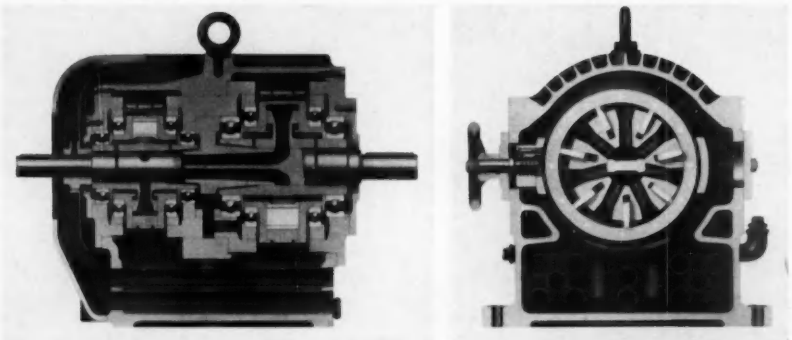


Fig. 37. Sectional views of a Churchill-Sturm hydraulic drive.
(Courtesy of Churchill-Sturm Limited)

The housings are also mounted on ball bearings in their sliding frames and are free to rotate in phase with the rotors. A cooling fan is mounted on the input shaft, blowing air over fins on the top of the outer casing and through tubes set in the oil reservoir.

The pump forces oil through one channel of the central shaft, driving the motor, and the oil discharged by the motor returns through the other channel of the central shaft to the pump. Part of the oil in circulation is continuously bled off to lubricate the bearings and to seal the circuit against infiltration of air, oil for replacing the amount used in this way being drawn from the reservoir.

With the controls in the " stop " position, the pump housing is concentric with the rotor, so that there is no displacement of oil, i.e., the motor remains at rest. When the control is operated, the housing is moved into an eccentric position, oil begins to flow to the motor, the housing of which is then at maximum eccentricity, and the motor starts to rotate. By the time that the pump housing reaches its maximum eccentricity, the motor speed has risen to about 20% of its maximum rating, and the motor is developing its full horsepower. During this period, the torque, which depends directly on the eccentricity of the motor housing, remains constant. From this point onward, the pump housing remains at maximum eccentricity, so that the volume of oil delivered to the motor, and hence the power developed, remain constant. The speed of the output shaft can be increased by reducing the eccentricity of the motor housing, torque falling off proportionately, but power output remaining unchanged. Maximum speed is reached when the eccentricity of the motor housing is reduced to about 20%, as shown in Fig. 38, which gives the relationship between the eccentricities of the pump and motor housings and the speed of the output shaft.

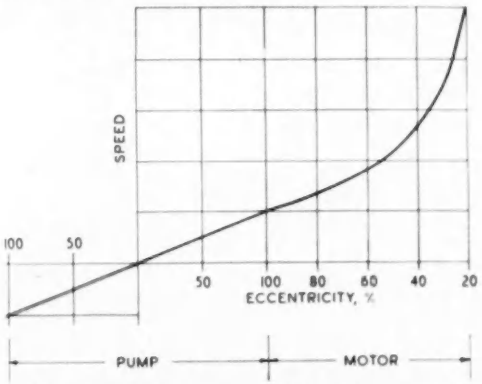


Fig. 38. **Relationship between the eccentricities of the pump and motor housings of the Churchill-Sturm drive and the speed of the output shaft.** *(Courtesy of Churchill-Sturm Limited)*

Thus, with the oil pump driven at constant speed, the speed of the motor can be infinitely varied from zero to maximum in either direction, first by increasing the eccentricity of the pump housing, and then by reducing the eccentricity of the motor housing. With a constant oil pressure, the torque remains constant up to about 20% of maximum speed, by which time the power transmitted has risen from zero to maximum; thereafter, the horsepower remains constant throughout the rest of the speed range, while torque falls off proportionately with increasing speed. Efficiency is virtually constant at all normal operating speeds, varying from about 68 to 72% at half load, and from about 75 to 80% at full load.

Reversal of the direction of rotation of the output shaft is accomplished by moving the control in the opposite direction, thus changing the eccentricity of the pump housing relative to the rotor from one side to the other, and reversing the direction of oil circulation.

The Churchill-Sturm hydraulic drive is available in various sizes, with output capacities ranging from about 1½ hp at 2000 rpm up to about 32 hp at 750 rpm. Various types of reduction gears are available for application to the drive, if lower speed ranges are required. A special type of Churchill-Sturm drive is a vertical unit with a flange-mounted electric motor. This type, which is available in three sizes, i.e., 1½, 3½, and 7 hp, constitutes a compact, self-contained drive for use when a vertical shaft is required.

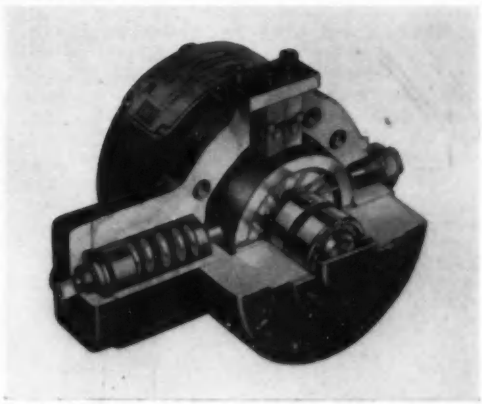


Fig. 39. **Cut-away view of a Racine model " Q " 5-gpm variable-volume pump with automatic pressure-compensating control.** *(Courtesy of Racine Hydraulics & Machinery, Inc.)*

The Racine variable-volume vane-type pumps, made in the U.S.A. by Racine Hydraulics & Machinery, Inc., are unique in that they incorporate an automatic pressure-compensating control which can be preset to maintain a given pressure and which permits delivery of only enough fluid to compensate for leakage losses in a system and which adjusts delivery automatically in accordance with load requirements.

This pressure-compensating feature is a source of force transmitted by a spring, as shown on the left of Fig. 39, or hydraulically, and serves to hold the pump pressure ring in its full-delivery position. When the system pressure creates more force in the ring than the spring force, it will cause the pressure ring to move to the left, compressing the spring. Thus, the pressure ring will be moving towards the neutral or no-delivery position, and the two opposing forces will balance at a point where just enough delivery is produced to maintain system requirements. Similarly, any drop in system pressure causes the ring to move in the opposite direction, i.e., to a position for greater pump delivery. In this way, by ensuring that only sufficient fluid for effective operation of the hydraulic system is delivered by the pump, power is saved and efficiency is accordingly increased. An adjustable end-stop (right of illustration) is provided for setting maximum delivery.

These pumps, which are of the unbalanced type, are fitted, as shown in the illustration, with slanted vane slots,

and are non-reversible. They are available in a range of sizes and alternative methods of mounting, with capacities ranging from about 5 gpm at 1000 psi and 1800 rpm up to about 70 gpm at 1000 psi and 1200 rpm. Corresponding hydraulic motors, also of vane type but of fixed displacement, and with non-slanting vane slots, permitting reversibility, are available in a range of five sizes, with a maximum capacity of about 45 hp.

If desired, the automatic pressure-compensating control on the pump can be removed and replaced by a manual or other suitable control for effecting arbitrary delivery settings, in the same way as a conventional variable-delivery pump. Usually, however, the automatic pressure-compensating control is retained, and infinitely variable speed is achieved by means of a suitable control valve, as already mentioned in the section dealing with fixed-displacement pumps and motors.

In addition to their variable-delivery axial-piston pumps and fixed-displacement axial-piston motors, already mentioned, the Denison Engineering Division of the American Brake Shoe Company also produce a range of variable-delivery vane-type pumps and fixed-displacement vane-type motors. These pump units are available in three basic sizes, i.e., 5, 15, and 25 gpm at 1200 rpm and 1000 psi for continuous duty. The motor units (which can also be used as fixed-displacement pumps) are available with interchangeable cam rings, providing a high degree of adaptability, with maximum power outputs from about 2 to 45 hp at 1200 rpm.

Special Types

In dealing with the hydraulic units previously described, no difficulties have been experienced in classifying them in one or another category. However, there are one or two types which do not lend themselves to ready classification in this way. Thus, the " Deri Sine " and " Keelavite " units described below may be said to incorporate vaguely some of the features of vane types, or even of gear types, while the " Keelavite ", in fact, actually employs its own designation of " rotary-abutment " unit. For these reasons, it appears preferable not to complicate matters by including them in a category which is not strictly applicable, but to list them as special types.

" Deri Sine " Hydraulic Units:—

Although available both as a fixed-displacement pump and a fixed-displacement motor, for infinitely variable speed applications the " Deri Sine " unit, made in the U.K. by Andrew Fraser & Co. Ltd., can be used as a motor in conjunction with, among others, the " Mono-Radial " variable-displacement radial-piston pump, already described. The Deri Sine hydraulic motor (Fig. 40) basically consists of two double-cam rotors (1) separated by a stationary dividing plate (2). The two rotors are keyed to the output shaft (4), the apexes of each being set at right angles to the other, and rotate inside concentric stator chambers (3), each of which is divided into two pumping chambers by means of brushes (5) located in radial slots in the stators. The brushes (5) are held in initial contact with the rotors (1) by means of a spring-loaded rocker (6) pivoted in the dividing plate (2), and thereafter by hydraulic pressure, only a small differential area between the top and bottom faces being effective. There are thus two equally spaced pumping chambers swept by each rotor, the sum of the capacities of both pairs of chambers being at all times constant. No valves are used, suction and discharge being accommodated by simple inlet ports (7) and outlet ports (8).

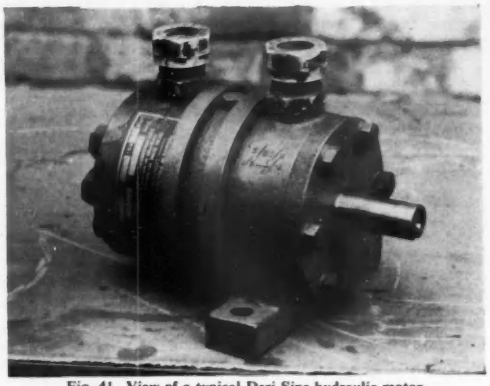
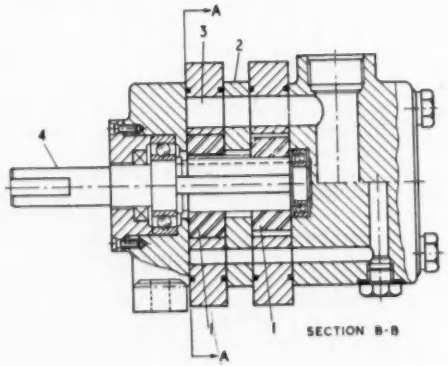


Fig. 41. View of a typical Deri Sine hydraulic motor.
(*Courtesy of Andrew Fraser & Co., Ltd.*)

The Deri Sine motor is capable of rotation in either direction by reversal of a simple control valve. Full starting torque is transmitted and constant torque maintained throughout the speed range. These motors are available as foot- or flange-mounting models, with or without double shaft extension, in a variety of sizes, with outputs ranging from about 7 hp at 1000 rpm to about 90 hp at 1000 rpm. A view of a typical " Deri Sine " motor is shown in Fig. 41.

" Keelavite " Rotary-Abutment Hydraulic Units:—

" Keelavite " rotary-abutment units, made in the U.K. by Keelavite Rotary Pumps & Motors Ltd., form a range



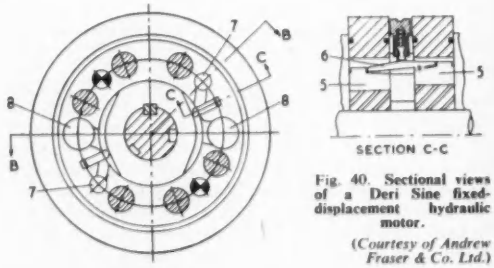


Fig. 40. Sectional views of a Deri Sine fixed-displacement hydraulic motor.
(*Courtesy of Andrew Fraser & Co. Ltd.*)

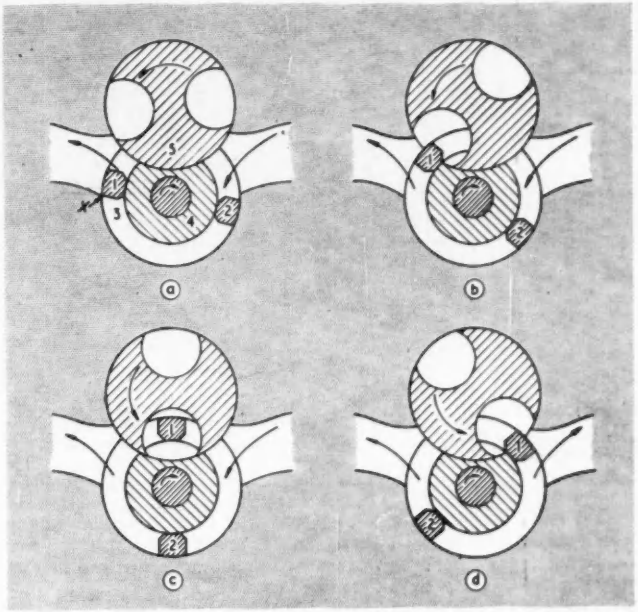


Fig. 42. **Diagrammatic representation of the rotating components of a Keelavite rotary-abutment hydraulic unit at different angular positions during half a revolution.**

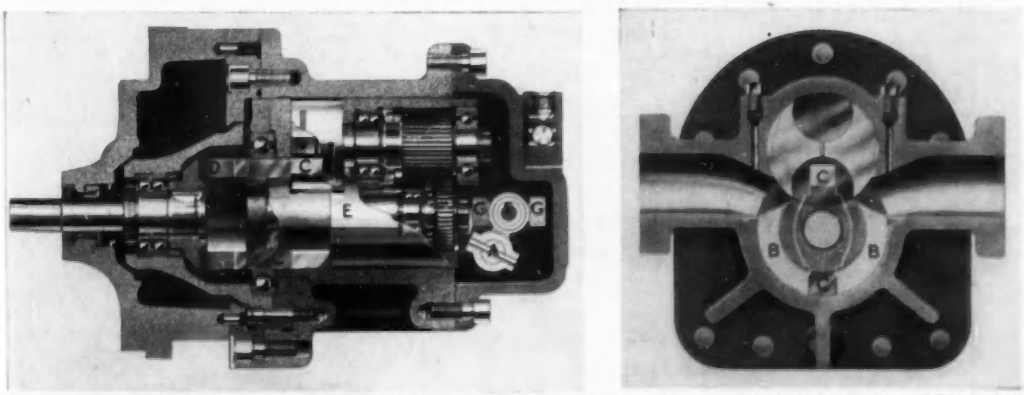
of hydraulic pumps and motors with either variable or fixed capacity, depending on requirements. The general principle on which they operate is, however, the same in all cases, i.e., that of two rotating members contained in a casing. One of the two rotating members (Fig. 42), i.e., the rotor, is the pumping member and from it project axially two blades (1) and (2), moving in the annular space (3) about the fixed central portion (4). The other rotating member is the rotary abutment (5), the function of which is to maintain a seal on one side of the annular space (3), between the inlet and outlet ports.

The rotary abutment is accommodated in a recess of circular form in the fixed central portion (4) and is provided with two longitudinal recesses which receive and shroud the blades (1) and (2) as they pass through that part of the annulus lying between the inlet and outlet ports. There is no contact between the rotary abutment and the rotor, the correct angular relationship between these components being maintained by timing gears.

The chord of the recess in the fixed central portion (4) is greater than the width of the recesses in the rotary abutment (5), so that an area seal is always maintained between the inlet and outlet ports. The timing need not be accurate and, since torque is not transmitted by the abutment, the loads on the teeth of the timing gears are very light.

Fig. 42 shows diagrammatically the rotating components at different angular positions during half a revolution. In Fig. 42a, at the instant when blade (1) reaches point *X*, the other blade (2) commences to deliver fluid, while blade (1) passes into the recess in the rotary abutment (Figs. 42b and 42c), merely displacing its own volume of fluid as it moves. During the time interval represented by Figs. 42a to 42c inclusive, blade (2) has been displacing fluid at a constant rate around the annulus (3), and this condition also continues during the condition shown in 42d, in which blade (1) is now passing out of the recess in the abutment, thus permitting fluid to enter from the inlet port and fill the space. From the position shown in Fig. 42d, the conditions in Fig. 42a are once more reached, except that blade (2) is now taking the place of blade (1). As a result, it will be appreciated that, because no change occurs in the rate of delivery throughout the cycle, the output is entirely free from pulsation.

A sectional and transverse view of a Keelavite variable-capacity unit is shown in Fig. 43. The rotor blades *C* engage with axial slots in the rotating housing *D*, by means of which the rotor is driven. At the end of the drive shaft remote from the rotor are the timing gears for driving the rotary abutment. A non-rotating centre *E*, through which runs the rotor shaft, provides a movable end wall to the annular pumping space *B* and, by moving axially with the rotor assembly, forms a continuous seal with the ends of the rotor blades. The other end of the



ig. 43. **Sectional and transverse views of a variable-capacity Keelavite rotary-abutment unit.** (*Courtesy of Keelavite Rotary Pumps & Motors Ltd.*)

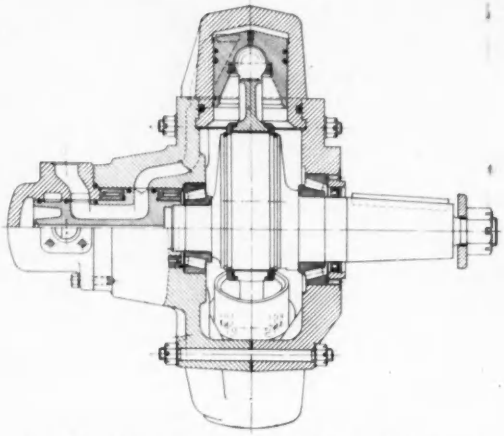


Fig. 44. **Section of a " Staffa " radial-type low-speed hydraulic motor.** (*Courtesy of Chamberlain Industries Limited*)

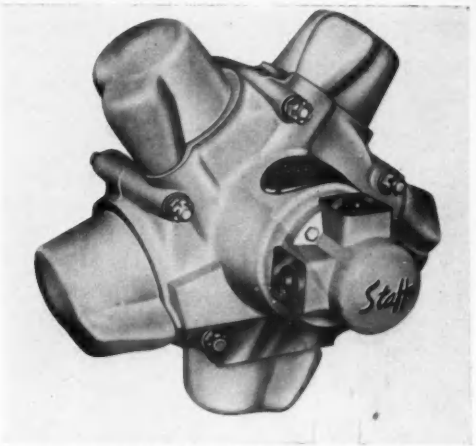


Fig. 45. **View of a " Staffa " radial-type low-speed hydraulic motor.** (*Courtesy of Chamberlain Industries Limited*)

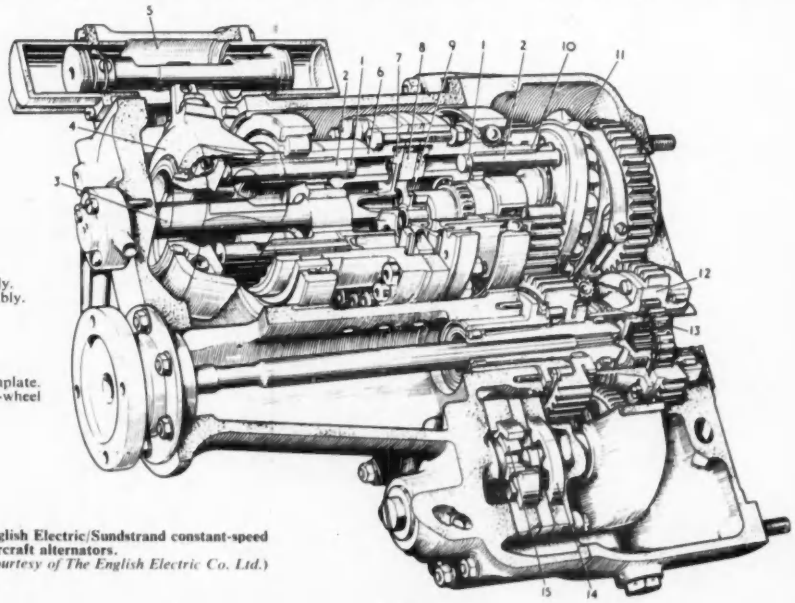
annular space is, of course, provided by the cylindrical end wall of the rotating housing *D*.

Axial movement is imparted to both the rotor and the rotor centre *E* by angular movement of the transverse shaft *A*, through the medium of a short lever carrying a self-aligning bearing *F*. The latter is thus caused to move freely between the forks *G*, which consequently transmit the required axial movement to the rotor assembly. The capacity of the unit is varied by axial movement of the rotor assembly, controlling the effective axial length of the annular pumping space *B* and hence the actual volume swept during each revolution of the rotor blades.

All Keelavite rotary-abutment units can function in either direction with equal efficiency. In the case of a motor unit, reversing is effected by the operation of an external valve which interchanges inlet and outlet. This procedure can take place at any point during a revolution, irrespective of the relative positions of the pumping elements. Keelavite rotary-abutment variable-speed units are at present available in only one size with a maximum power rating of about 15 hp at 1500 rpm. Various sizes of fixed-displacement pumps and motors are, however, produced.

Combinations of Variable-Delivery Pumps and Motors from Different Manufacturers

While it is true that the majority of manufacturers of variable-delivery pumps also produce the corresponding range of fixed- and/or variable-displacement motors for operation in conjunction with their pumps, some do not. Similarly, there are a few manufacturers who do not produce pump units but have developed fixed-displace-



(1) Pistons.
(2) Push rods.
(3) Manifold assembly.
(4) Pump swashplate assembly.
(5) Swashplate control assembly.
(6) Pump cylinder block.
(7) Pump valve plate.
(8) Port plate.
(9) Motor valve plate.
(10) Motor cylinder block.
(11) Motor (fixed-angle) swashplate.
(12) Output gear and free-wheel clutch assembly.
(13) Input gear.
(14) Charge pump.
(15) Scavenge pump.

Fig. 46. **Schematic view of English Electric/Sundstrand constant-speed drive for aircraft alternators.** (*Courtesy of The English Electric Co. Ltd.*)

ment motor units only (mainly for special applications), for use with a suitable variable-delivery pump unit from some other manufacturer.

There is, of course, no fundamental objection to this procedure, provided that the comparative characteristics of the two units are suitably matched. For this reason, close co-operation between the manufacturers concerned is essential.

An example of a fixed-displacement motor intended for operation in conjunction with a suitable variable-delivery pump unit from another manufacturer is shown in Figs. 44 and 45. This motor, made in the U.K. by Chamberlain Industries Limited, is a radial-piston type with an eccentric on the drive shaft and is a high-torque, slow-speed unit, developing about 30 hp at a maximum speed of 50 rpm and 1500 psi.

Constant Output-Speed Applications

In discussing hydrostatic drives in general, this survey has so far been concerned with normal applications involving a constant input speed and a variable output speed. It should, however, be appreciated that these drives, by means of suitable control arrangements, can also be used in applications where it is desired to maintain a virtually constant output speed with variable-speed prime movers. Examples of such applications include transmissions for refrigerated trucks, in which the refrigeration compressor is driven via a power take-off from the truck engine, and constant-speed alternator drives for aircraft electrical systems. Several manufacturers have developed suitable drives for applications of this nature. Thus, for example, a hydraulic constant-speed drive for refrigerated trucks is available in the U.S.A. from the Sundstrand Machine Tool Co., while package-type units for constant-speed alternators in aircraft have been developed in the U.K. by The English Electric Co. Ltd. and by H. M. Hobson, Ltd.

Basically, the English Electric system employs a Sundstrand axial-piston variable-delivery pump and a fixed-delivery motor, combined in one cylinder assembly and arranged in such a way that only the power represented by the difference in input and output speeds is handled by the pump and motor units. Several types have been developed, a typical English Electric/Sundstrand constant-speed drive being shown in Fig. 46, which refers to a 40-kVA unit designed to transmit 50 hp at a constant speed of 6000 rpm over an input-speed range of 3000 to 8300 rpm.

In this design, when the pump swashplate is in its maximum angular position, the output shaft is overspeeding the input shaft. After gear-ratio corrections, this overspeed is the ratio of maximum pump displacement to the fixed displacement of the motor minus piston leakage. The swashplate remains in this maximum stroke position until the output speed reaches 6000 rpm. With a further increase in input speed, the normal tendency is to increase the output speed. At this point, however, the governor regulates the control piston of the swashplate control assembly (5) to decrease pump-piston stroke. Less oil then flows from the pump to the motor, and therefore the differential speed between the motor shaft and the cylinder-block assembly is reduced to retain constant output speed.

When the input speed exceeds the output speed, the cylinder-block assembly will be turning faster than the motor-shaft, which always operates at constant speed. Relative motion between the cylinder block and motor shaft will now be in a reverse direction, since it is now necessary to subtract from input speed to maintain governed output speed. Owing to the cylinder block overspeeding the motor shaft, the pump swashplate is held in an underdrive position (negative angle), permitting an oil flow from motor to pump. In effect, the pump is now motoring, and the power flow in the hydraulic units is reversed.

The alternator is driven through a full-complement, over-running, sprag-type, one-way clutch located in the output gear and clutch assembly (12) of the transmission. This clutch is necessary to prevent power from being directed through the transmission opposite to that from the main aircraft engine when the alternator is tending to act as a motor.

As the output speed has to be regulated to 6000 rpm constant, the speed signal for the governor system is taken from the output gear and clutch assembly. This signal is transmitted through the tachometer drive shaft and a centrifugal switch to the governor assembly. A charge-pump picks up oil from the external reservoir and delivers it through the cylinder-block assembly, whence it returns through a manifold return line to a charge-pressure relief valve, which maintains the charge pressure at 300 psi. A by-pass taps off oil from the manifold return line and supplies oil to the governor system and swashplate control pistons.

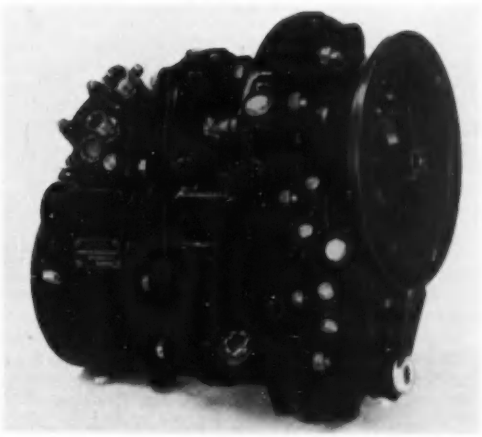


Fig. 47. View of Hobson constant-speed unit for aircraft alternators. *(Courtesy of H. M. Hobson, Ltd.)*

A view of the Hobson type-114 constant-speed unit for aircraft alternators is shown in Fig. 47. This unit, which is intended for driving a 10-kVA alternator and is capable of maintaining a constant power output of 15 hp at 8000 rpm over an input-speed range of 3300 to 10,000 rpm, with a tolerance on output speed of $\pm 1\%$, also incorporates an axial-piston variable-delivery pump and a fixed-delivery motor. Control of the pump swashplate is achieved by a governor-controlled servo-mechanism which comprises a pair of spring-loaded fly-weights on the output shaft, a relay valve, and a servo-piston mechanically connected to the swashplate. In this way, the rate of delivery to the hydraulic motor can be increased or decreased, thereby causing it to add to or subtract from the rotational speed of the input shaft.

HYDROKINETIC DRIVES

Basically, all hydrokinetic drives (or hydrodynamic drives, as they are sometimes called) utilize an impeller or pump member connected to the drive shaft and imparting kinetic energy to a working fluid under closed-cycle conditions, transferring the energy to a turbine member connected to the driven shaft.

Hydrokinetic drives may be divided into three types, i.e., hydraulic couplings, torque converters, and converter-couplings, and these will be discussed separately. It is interesting to note, however, that hydrokinetic couplings and converters stem from the principles developed by Dr. Föttinger and embodied in his " Transformator ", which was essentially a torque converter. Between 1907 and 1912, these drives were principally used for marine applications in the German navy, some of them being capable of transmitting up to 35,000 hp. Owing, however, to their limited efficiency, they were subsequently superseded by the more efficient Parsons type of mechanical reduction gear.

In the meantime, Föttinger's hydraulic coupling, developed in a more efficient form by Dr. Bauer at the Vulcan Works, Hamburg, in conjunction with helical reduction gearing, was being successfully applied to diesel-driven ships and enjoyed considerable popularity after World War I. Similarly, the Vulcan coupling found considerable application in steam reciprocating engines and in exhaust turbines, under the name of the Bauer-Wach system.

However, despite its acceptance in marine propulsion, no attempts were made to use the Vulcan coupling in industry until 1926, when H. Sinclair realized its potentialities as a means of improving automotive power transmissions. The result was the development of a greatly simplified form of coupling, the Vulcan-Sinclair fluid coupling, for industrial and automotive drives. In 1930, the Daimler Company adopted this coupling in England, in conjunction with a planetary gearbox, under the well-known name of the fluid-flywheel transmission. This form of transmission was shortly afterwards standardized in all Daimler and Lanchester motor cars and also in Daimler and A.E.C. buses. It is still the type of transmission largely standardized in buses. It is also of interest to note that the Chrysler Corporation in 1938 introduced the first American automobile with a fluid coupling constructed under Vulcan-Sinclair patents, under the name of " Fluid-Drive ", and several million of these drives were incorporated in Chrysler, de Soto, and Dodge cars.

Interest in the torque converter revived in the 1920's, particularly with regard to automotive applications, and several important types were developed, including the Vickers-Coats converter and the Lysholm-Smith converter. Now that efficient types of both torque converters and couplings were actual realities, it was inevitable that attempts would be made to produce units combining the advantages of both types. Thus, in 1930, P. Salerni patented converter-coupling designs which were subsequently developed into the Salerni Transmitter manufactured in England by Brockhouse Engineering Ltd. in 1941. Similar developments in this category included the Trilok (Germany, 1934) and the Schneider Brothers (U.S.A., 1940).

As a result of the success of the Lysholm-Smith converter and the Schneider converter-coupling in tanks, etc. during World War II, the White " Hydro-Torque " was introduced in 1946 for buses in the U.S.A. Since then, development in America of transmission systems involving the use of some form of torque converter or converter-coupling, usually in conjunction with a self-changing gearbox, has been rapid, starting with the Buick " Dynaflow " in 1948 and the Packard " Ultramatic " in 1949. By 1951, nearly all American automobile manufacturers were utilizing devices of this kind for their transmissions, including Chrysler, who decided to adopt converter-couplings in place of the hydraulic couplings which, as already mentioned, they had previously been using. To-day, many automobile manufacturers produce their own hydraulic transmission drives but, as these are mainly intended for exclusive application to particular makes of car and are not inherently designed for industrial applications, details of automatic transmissions for motor cars are not included in this survey.

Hydraulic Couplings

The hydraulic coupling is the simplest form of hydraulic drive, consisting essentially of a pump member or impeller (2), combined with a casing (4) connected to the input shaft (1), and a turbine member or runner (5) connected to the output shaft (7), as shown in Fig. 48. The impeller and turbine members comprise saucer-shaped discs fitted with flat radial vanes for directing fluid flow, different numbers of vanes usually being provided in the impeller and the turbine, to counteract the effects of vane interference and the consequent possibility of irregular output torque. To assist in establishing fluid circulation, a guide ring (3) is sometimes fitted. In addition, an anti-drag baffle (6) may be incorporated.

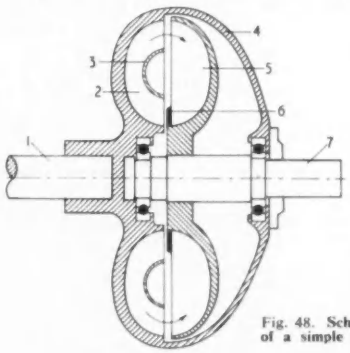
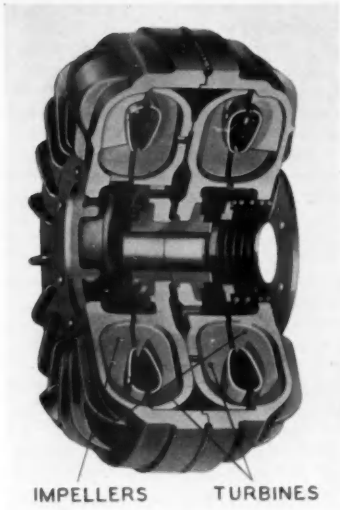


Fig. 48. Schematic arrangement of a simple hydraulic coupling.

Power is transmitted by the circulation of oil between the two members, an oil reservoir being formed by the casing (3) when the impeller is stationary. When the impeller is rotated, centrifugal action causes an outward flow of the oil, and the application of torque to the system gives rise to a slight change in the relative speeds of rotation, initiating fluid flow.

In one design of hydraulic coupling, made in the U.S.A. by the Twin Disc Clutch Company, use is made of twin circuits, each comprising one impeller and one turbine, thus balancing any effects of axial thrust. The basic components and an assembled clutch of this type are shown in Fig. 49.

It should be appreciated that the quantity of oil flowing at any period in a hydraulic coupling is identical



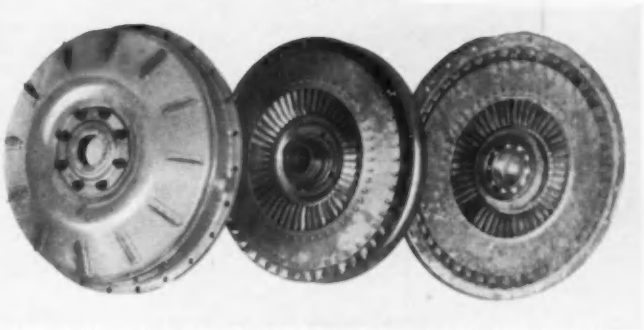


Fig. 49. Cut-away view of a " Twin Disc " hydraulic coupling and its basic components, showing the two impeller units and (centre) the two combined turbine units. (*Courtesy of Twin Disc Clutch Company*)

for both the impeller and the turbine. Torque conditions are therefore always identical on both members. In other words, input torque must equal output torque and there can be no torque conversion, so that there can be no increase in torque on the turbine element over and above that received by the impeller from the input shaft. Any coupling losses take the form of slip, i.e., a difference between impeller and turbine speeds, so that the efficiency of a hydraulic coupling can generally be expressed in terms of the ratio of output to input speed. The presence of slip is naturally accompanied by the generation of heat, but, as the normal slip of a hydraulic coupling of the type described is only from about 2 to 4%, temperature rise is slight and heat dissipation from the coupling is easily taken care of, without the need for special cooling equipment.

It should also be noted that torque transmission with a hydraulic coupling would not be possible, were it not for the presence of slip. As the result of centrifugal force, the fluid at the periphery of the impeller is subjected to a certain pressure, causing it to circulate through the turbine until, as a result of increase in turbine speed, a counter-centrifugal force is set up, tending to stop oil circulation. Hence, if the speeds of the impeller and the turbine were equal, circulation of oil would cease entirely, and no torque would be transmitted.

Both the torque and output speed of a hydraulic coupling (and, for that matter, of a torque converter) are affected by input speed, the torque transmitted varying as the square of input speed, and output speed linearly with input speed.

Hydraulic couplings of this type are deservedly popular in many applications and are constructed by several manufacturers for traction or other purposes, e.g., in the U.K. by the Fluidrive Engineering Co. Ltd. (the Vulcan-Sinclair coupling) and by Crofts (Engineers) Ltd., in the U.S.A. by the American Blower Co. and by the Twin Disc Clutch Co., and in Germany by J. M. Voith G.m.b.H. It will be appreciated, however, that they are essentially constant-speed units in that no provision is made for their use as a variable-speed drive. In fact, they may be considered as hydraulic clutches and, as such, are outside the scope of this survey.

An infinitely variable speed hydraulic coupling embodying what is termed scoop control is shown schematically in Fig. 50. In this type of coupling, an outer casing (1), integral with the impeller (5) and rotated by the input shaft (3), surrounds the casing in which the impeller (5) and turbine (4) rotate. By means of a number of leak-off nozzles (2), oil is forced centrifugally from the inner to the outer casing, which is of sufficient capacity to be able to contain the entire oil charge in the form of an annulus. A scoop tube (6), adjustable by a lever (8), dips into this belt of oil, so that oil can be ducted via an axial channel back into the inner casing. Thus, with the scoop tube in its lowest position, the quantity of oil in circulation is at its maximum ; if, however, the tube is raised, the quantity of oil in circulation is reduced, with an increase in slip, depending on the position of the tube. In this way, it is possible to vary the speed of the output shaft (7) over a wide range, depending on the torque characteristics of the driven machine.

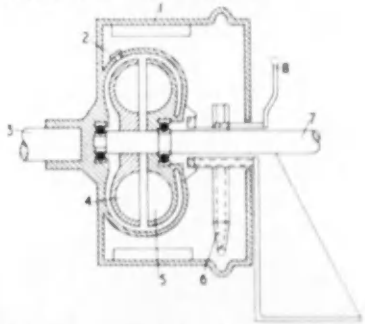


Fig. 50. Schematic arrangement of a variable-speed hydraulic coupling with scoop control.

It will, of course, be understood that any increase in slip is accompanied by a rise in oil temperature. In many instances, this is of minor importance, and no special provisions for oil cooling are necessary. If, however, overheating is a possibility, the design can incorporate an oil cooler.

In the design just described, the scoop tube is adjustable. In other types of scoop control, however, the scoop tube may be kept at a constant level, an auxiliary pump being used to remove oil from the circuit and to vary the amount in circulation.

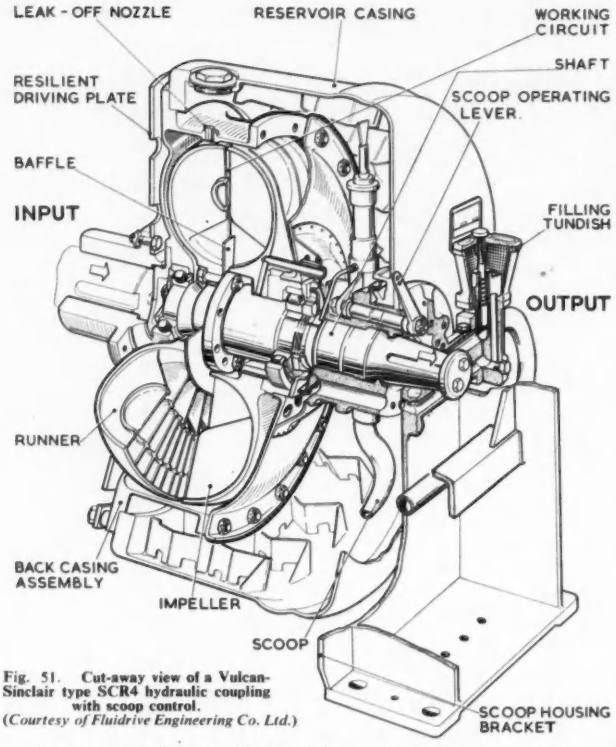


Fig. 51. Cut-away view of a Vulcan-Sinclair type SCR4 hydraulic coupling with scoop control.
(Courtesy of Fluidrive Engineering Co. Ltd.)

An example of a variable-speed hydraulic coupling with adjustable scoop control is the Type SCR4 Vulcan-Sinclair scoop-control coupling, made in the U.K. by the Fluidrive Engineering Co. Ltd. In this design, the major features of which are illustrated in Figs. 51 and 52, labyrinths with large clearances are used, since, when stationary, the oil is contained in the lower part of the reservoir casing. The use of these labyrinths, with their large clearances, eliminates frictional wear. These couplings are available in a wide range of sizes, with capacities from about 2 to 1500 hp at 1500 rpm.

An example of a variable-speed hydraulic-coupling unit, with adjustable scoop control and a cooling pump for oil is the Vulcan-Sinclair type ST shown in Fig. 53. Especially designed for operation with high-speed, high-efficiency squirrel-cage motors, and providing infinitely variable speed regulation for boiler-feed-pump drives, gas boosters, and similar applications, this design, which is also manufactured in England by Fluidrive, basically comprises a resilient-diaphragm driving disc with driving boss, a sliding-scoop hydraulic coupling, the output shaft of which is supported by a self-aligning bearing in the scoop housing bracket, and a small motor-driven centrifugal pump for oil circulation, mounted in the sump or reservoir tank. The pump draws oil from the reservoir tank and delivers it through water-operated oil coolers into the stationary scoop housing bracket. Variable-speed couplings of this type are made in a range of capacities up to about 1500 hp at 3000 rpm. In addition, a range of double-circuit couplings of the same basic type, designated DST couplings, is available, with capacities ranging from about 1500 hp to 20,000 hp at 3000 rpm.

Another representative Vulcan-Sinclair Fluidrive coupling with scoop control and cooling arrangements is the type SCD, specially developed for marine propulsion. This entirely self-contained design (Fig. 54), utilizes a large scoop tube for transferring oil under pressure to the working circuit, permitting fast filling, while rapid emptying is assured by means of quick-emptying centrifugal and diaphragm valves. The diaphragm valves open when the scoop tube is withdrawn, and the centrifugal valves when the engine speed is reduced, in both cases discharging the oil from the working circuit directly to the rotating reservoir. Oil is circulated continuously through leak-off nozzles into the rotating reservoir, being picked up by the scoop and returned to the circuit after passing through a water-operated oil cooler. When used with a reversing engine, a double-ended scoop tube is fitted, to pick up the oil in the reservoir in either direction of rotation, the trailing scoop being automatically sealed off. These couplings are available in a wide variety of sizes, with capacities from about 200 to about 4400 hp at 400 rpm.

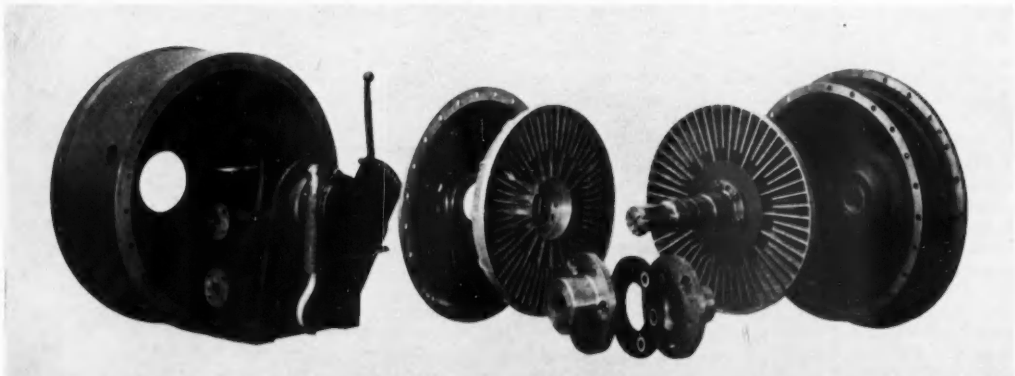


Fig. 52. Main components of a Vulcan-Sinclair type SCR4 hydraulic coupling with scoop control. *(Courtesy of Fluidrive Engineering Co. Ltd.)*

Vulcan-Sinclair variable-speed couplings are deservedly popular and are extensively used in a wide range of applications. In addition to being made in the U.K. by Fluidrive, they are also made under licence in other countries, including the U.S.A., France, Italy, Germany, and Spain.

An example of a variable-speed hydraulic coupling with a fixed scoop tube and an auxiliary pump for varying the amount of oil in circulation is shown overleaf in Fig. 55. In this design, which is one of a large variety of hydraulic couplings made in Germany by J. M. Voith G.m.b.H., oil is removed by the fixed scoop tube (1) and returns to the system after passing through the cooler (2). By means of a pump (3), the total volume of oil in the circuit can be increased or reduced, as required, supplies of oil being drawn from or discharged into the reservoir (4), depending on the direction of rotation of the pump (3). Couplings of this type, designated SL, are available in several sizes, with a maximum capacity of about 3500 hp at 1450 rpm.

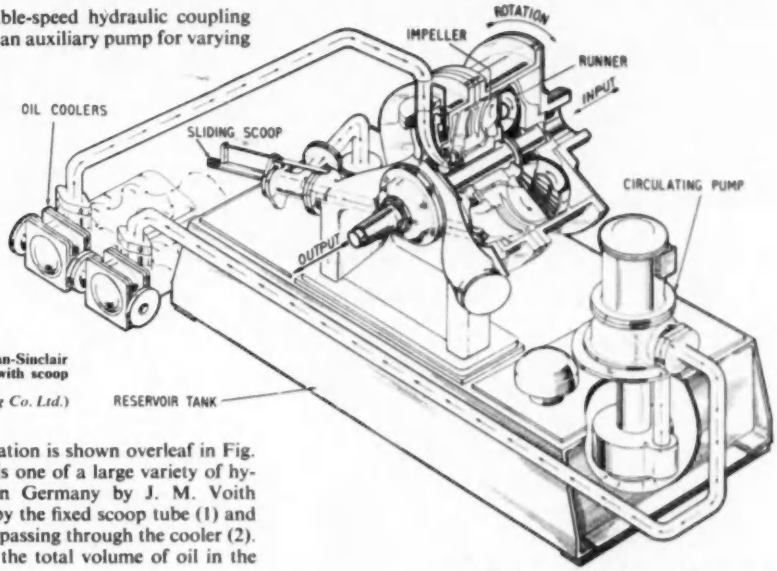


Fig. 53. Cut-away view of a Vulcan-Sinclair type ST hydraulic coupling unit, with scoop control and oil cooling.
(*Courtesy of Fluidrive Engineering Co. Ltd.*)

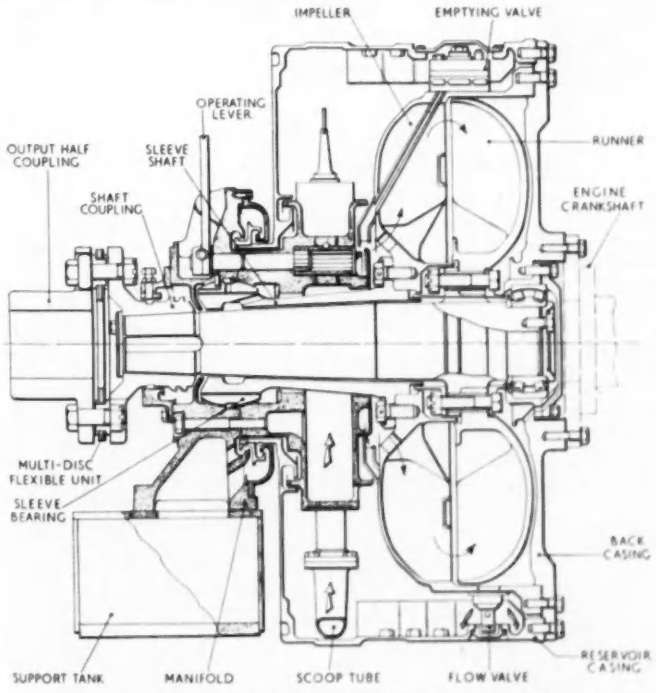


Fig. 54. Section through a Vulcan-Sinclair type SCD hydraulic coupling with scoop control, for marine propulsion.
(*Courtesy of Fluidrive Engineering Co. Ltd.*)

Torque Converters

In the case of a hydraulic coupling, if the turbine is overloaded, not only does its speed reduce but the prime mover also becomes overloaded and its speed is also proportionately reduced. To prevent this interaction, it is therefore necessary to avoid the close proximity of the impeller and the turbine. In the torque converter, this is effected by the introduction of a reaction member between these two units, this member consisting of a set of radial blades which are inclined at an angle to the axis of rotation and which are fixed to a frame, so that they themselves are not free to rotate. In operation, the impeller drives the turbine, the oil being returned to the impeller via the reaction member. The basic components of a typical torque converter are illustrated in Fig. 56 and show (left to right) the driving shell or casing, the turbine, the reaction member, and the impeller.

In the torque converter shown in section in Fig. 57, while the torque of the turbine and hence of the output shaft is dependent on the

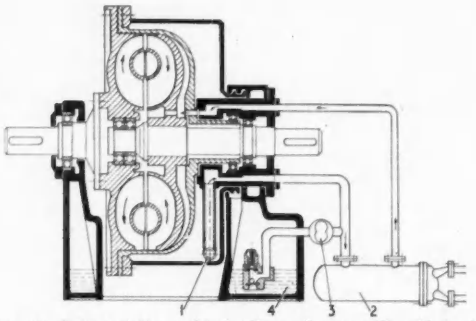


Fig. 55. **Voith variable-speed hydraulic coupling, type SL, with fixed scoop tube and control pump.**
(Courtesy of J. M. Voith G.m.b.H.)

change in fluid speed (in the direction of rotation of the input shaft) from *A* to *B*, the torque of the impeller depends on the difference in speed (still in the direction of input-shaft rotation) from *C* to *A* (see also Fig. 58). A change in the speed of the fluid in the direction of input-shaft rotation can thus be effected in the reaction zone from *B* to *C*, resulting in a torque being imposed on the reaction member.

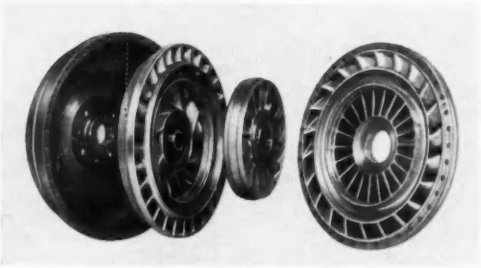


Fig. 56. **Basic components of a typical torque converter.**
(Courtesy of Brockhouse Engineering Ltd.)

If a hydraulic coupling is operating at maximum speeds and the turbine element is loaded so as to reduce its speed to about 50% that of the impeller, there is available a considerable pressure difference for inducing a large flow of oil around the circuit. As already mentioned, however, this large flow loads the impeller and slows down the prime mover as well. This is just where the reaction member does useful work in the torque converter. Thus, in a torque converter with the turbine loaded down to half speed, the flow of oil induced in the circuit is used to form a jet which drives the impeller forward, so that the impeller is not only being driven by the prime mover but also by a jet of oil streaming from the reaction member. Consequently, the more the turbine is slowed down by the load, the greater is the stream of oil forced through the reaction member, driving the impeller forward and maintaining engine speed.

Typical reaction vanes, with a steep inclination towards the direction of engine rotation, are shown in Fig. 58. In this connection, it should be appreciated that there is very little advantage to be gained by using anything but plain flat vanes in the case of a hydraulic coupling. If, for instance, the turbine blades of a hydraulic coupling are turned backwards against the direction of rotation, a higher torque can be absorbed by the turbine for the same quantity of oil circulating. However, owing to the exactly equal but opposite relationship always existing between the turbine and impeller of hydraulic couplings, any extra torque imposed on the turbine will also be imposed on the input shaft and will tend to stall the prime mover, so that nothing is gained by vanes of complex configuration except to provide a tortuous passage through which the oil must pass. In the case of the torque converter, however, the position is quite different, inasmuch as the reaction member breaks up this interdependence of turbine and impeller and permits the blades to be shaped in a variety of ways for producing a wide range of characteristics.

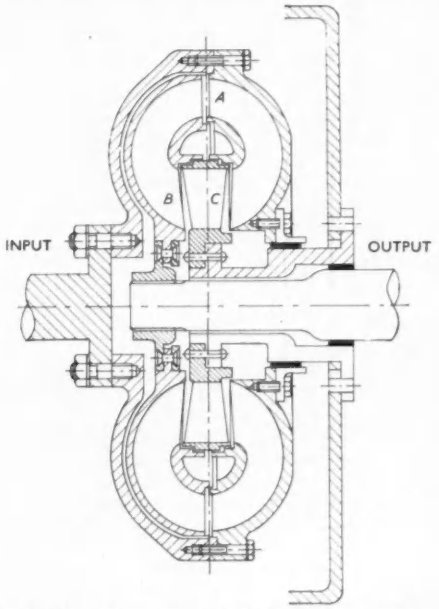


Fig. 57. **Schematic arrangement of a single-stage torque converter.**

For a given set of blade angles, a torque converter will have maximum efficiency at one set of fluid-flow conditions, i.e., at the design point, which is related to a particular ratio of output to input speed. In Fig. 59, which shows the performance data of a typical torque converter with constant input torque, the design point *A*, representing maximum efficiency, corresponds to a

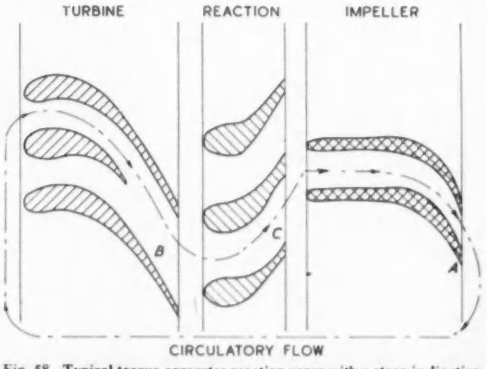


Fig. 58. **Typical torque-converter reaction vanes with a steep inclination towards the direction of input-shaft rotation.**

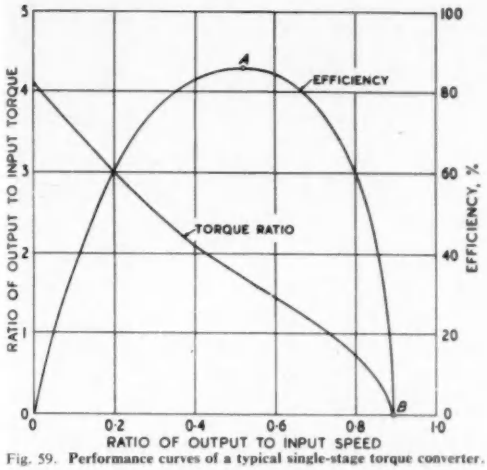


Fig. 59. **Performance curves of a typical single-stage torque converter.**

speed ratio of about 0·5. For all other speed ratios, there is a drop in efficiency, so that, as shown, the efficiency curve is roughly a parabola with zero points at zero turbine speed and also at the racing or run-away speed ratio (point *B*), when there is little or no relative speed between the fluid and the turbine blades and no output torque is therefore developed. The speed ratio at which racing occurs naturally depends on impeller and turbine design, but in most modern torque converters occurs at a speed ratio of slightly less than unity. Insofar as torque ratio is concerned, the maximum value of torque conversion occurs at zero turbine speed, dropping to zero at racing point. In simple single-stage torque converters of the type shown in Fig. 57, the maximum value of torque conversion, i.e., the stall torque ratio, usually has a value not exceeding 4:1.

So far, discussion has been limited to single-stage torque converters. Multi-stage converters are, however, quite common. In the multi-stage converter, use is made of two or more turbine members separated by corresponding reaction members. Thus, in a three-stage converter, the assembly consists of the impeller, a row of turbine blades, a row of reaction blades, a second row of turbine blades, another row of reaction blades, and finally a third row of turbine blades, the turbine assembly being connected to the output shaft.

The use of an increased number of stages gives an increase in torque-conversion ratio, among other advantages. However, fluid friction is often increased, so that the efficiency curve of a multi-stage converter often tends to fall off comparatively sharply, racing generally occurring at lower speed ratios than those for single-stage converters.

The basic elements of a three-stage torque converter made in the U.S.A. by the Twin Disc Clutch Company are illustrated in Fig. 60, which shows (left to right), the impeller, the three-row turbine assembly and the two-row reaction member. These units are shown assembled in Fig. 61, in which (1) is the impeller, (2) the turbine,

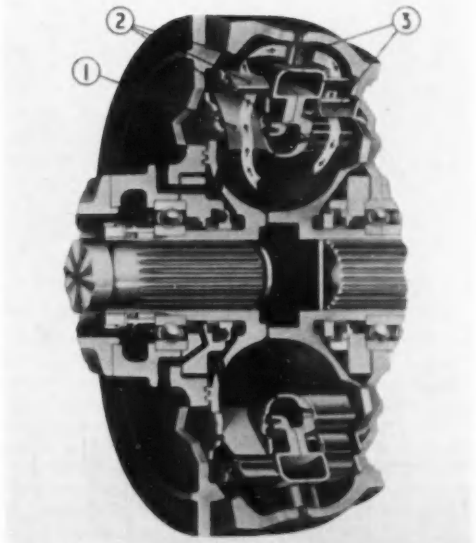


Fig. 61. **Cut-away view of assembly of the impeller, turbine, and reaction member in a " Twin Disc " three-stage torque converter.** *(Courtesy of Twin Disc Clutch Company)*

and (3) the reaction member. The impeller forces fluid outwards against the first ring of turbine blades. The fluid, having given initial rotation to the turbine, is then redirected by the first set of reaction blades and impinges against the second ring of turbine blades. The fluid is

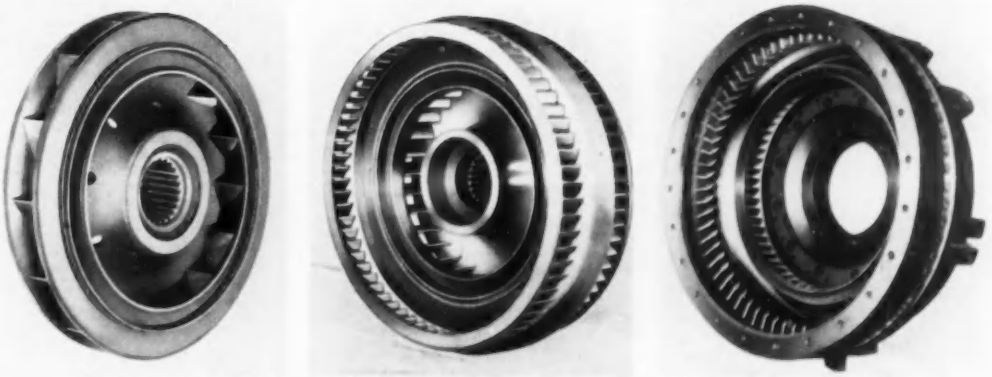


Fig. 60. **Basic components of a " Twin Disc " three-stage torque converter.** *(Courtesy of Twin Disc Clutch Company)*

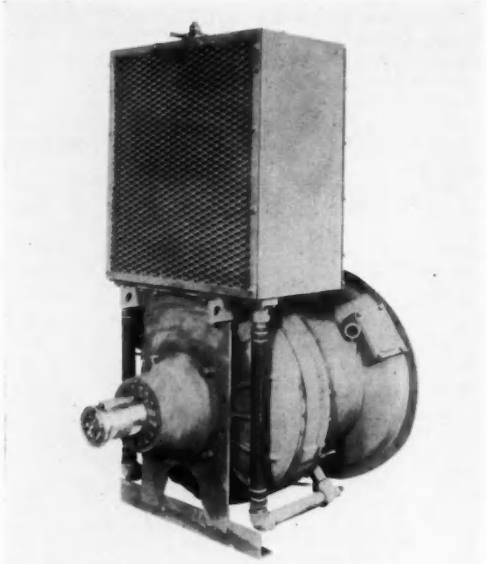


Fig. 62. View of " Twin Disc " heavy-duty three-stage torque converter, series 16,000, equipped with fan-cooled radiator. (*Courtesy of Twin Disc Clutch Company*)

directed and redirected in this way through all three sets of turbine blades. Under normal running conditions, the action of the impeller causes the turbine blades to turn almost freely, and the fluid passes through the converter easily and quickly, striking each blade at a very slight angle. When a load is encountered, the turbine slows down and the fluid then strikes the turbine blades at a sharper angle. As the force of the fluid is relayed from the first to the second to the third ring of turbine blades, the output torque accumulates until it is increased up to as much as six times the input torque.

As is the case with all torque converters and converter-couplings, some form of oil cooling must be provided. In the Twin Disc units, provision is made for circulating the oil through an external radiator, circulation being provided by the pressure differential across the impeller. The radiator may comprise a separate unit intended for mounting in any convenient position, or may be designed for direct mounting on the converter. Alternatively, cooling can be effected by means of a water-operated heat exchanger, this being the preferred arrangement.

These Twin Disc three-stage converters are available in various sizes, with maximum capacities ranging from about 330 hp at 2400 rpm to 1050 hp at 1700 rpm. A view of a Twin Disc heavy-duty three-stage torque converter with fan-cooled radiator, suitable for use with prime movers up to about 1000 hp and at speeds up to 1500 rpm, is shown in Fig. 62.

Fig. 63. View of " Twin Disc " series 1300 single-stage torque converter. (*Courtesy of British Twin Disc Ltd.*)

Single-stage torque converters are also produced by the Twin Disc Clutch Company. A feature of these units is that the fluid circuit differs from those normally used in other single-stage converters in that it is more or less pear-shaped, instead of round or elliptical. As a result, the fluid leaves the impeller blades at constant diameters. Two types are at present available, i.e., series 1500 and series 1300, the latter series being designed specifically to

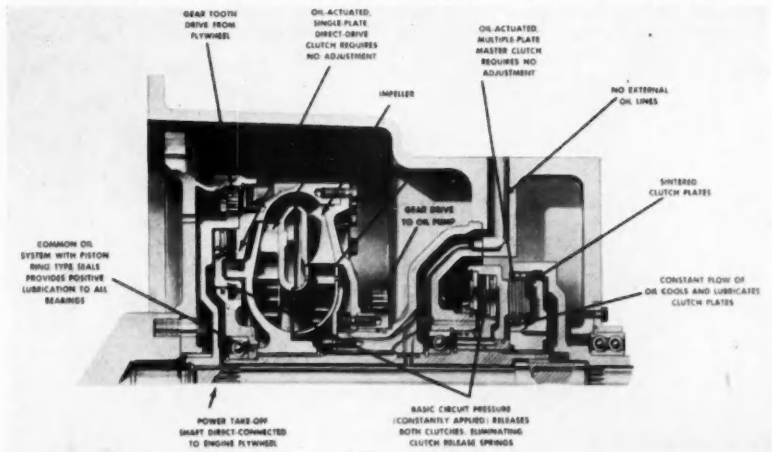

GEAR TOOTH DRIVE FROM FLYWHEEL
OIL-ACTUATED SINGLE-PLATE DIRECT-DRIVE CLUTCH REQUIRES NO ADJUSTMENT
OIL-ACTUATED MULTIPLE-PLATE MASTER CLUTCH REQUIRES NO ADJUSTMENT
IMPELLER
NO EXTERNAL OIL LINES
SINTERED CLUTCH PLATES
GEAR DRIVE TO OIL PUMP
COMMON OIL SYSTEM WITH PISTON RING TYPE SEALS PROVIDES POSITIVE LUBRICATION TO ALL BEARINGS
CONSTANT FLOW OF OIL COOLS AND LUBRICATES CLUTCH PLATES
POWER TAKE-OFF SHAFT DIRECT-CONNECTED TO ENGINE FLYWHEEL
BASIC CIRCUIT PRESSURE (CONSTANTLY APPLIED) RELEASES BOTH CLUTCHES ELIMINATING CLUTCH RELEASE SPRINGS

Fig. 64. " Case-o-Matic " drive for farm tractors, with single-stage Twin-Disc torque converter and built-in direct-drive feature. (*Courtesy of Twin Disc Clutch Co. and J.I. Case Co.*)

meet the requirements of modern high-speed diesel and petrol engines. The maximum capacity of the series 1500 converter is about 210 hp at 2500 rpm, and of the series 1300 (Fig. 63) about 215 hp at 3200 rpm. In both series, cooling is provided by a positive-displacement pump in conjunction with a simple heat exchanger, and the performance characteristics of each can be varied by alternative disposition of the circuit elements, so much so that it is possible, with these torque converters, to change the racing point from 0·85 to as much as 1·15.

As already stated, the range of Twin Disc single-stage torque converters is at present confined to the 1300 and 1500 series, i.e., to models with respective circuit diameters of 13 and 15 in. This range is, however, being extended and will shortly include the 1100 and 1800 series. In addition, the 1100, 1300, and 1500 series units will include special versions with a direct-drive feature, in which a friction clutch is incorporated (Fig. 64). This friction clutch is hydraulically operated by pressure oil delivered from an externally mounted gear pump, and serves to lock the turbine member (and hence the output shaft) with the input shaft, thus providing a direct drive at a 1:1 speed ratio whenever operating conditions are stabilized and there are no material torque fluctuations, e.g., in the case of a locomotive running at speed on the level, or a tractor running without load.

In the U.K., Twin Disc three-stage torque converters are manufactured under licence from the U.S.A. by Rolls Royce Ltd. for use with their own engines and also on behalf of British Twin Disc Limited for all other applications. In addition, British Twin Disc Limited are the manufacturers in the U.K. of the single-stage range of torque converters.

Converter-Couplings

As already stated, in the torque converter the reaction member is fixed to the frame and does not itself rotate. With this arrangement, there is only one point where engine and output torque are equal. Thus, in Fig. 65, which shows typical performance curves of a torque converter, the curve of efficiency reaches a maximum at the design point *A*, and then steadily decreases to zero at the racing point *B*. Similarly, the curve of torque ratio shows a decline, reaching its zero value at *B*. The point *O* on the torque-ratio curve, representing its intersection with the line corresponding to an output to input torque ratio of unity, is known as the coupling point or clutch point, and at speeds above this point the output torque is less than the input, so that something similar to overdrive conditions exists, during which efficiency falls off rapidly.

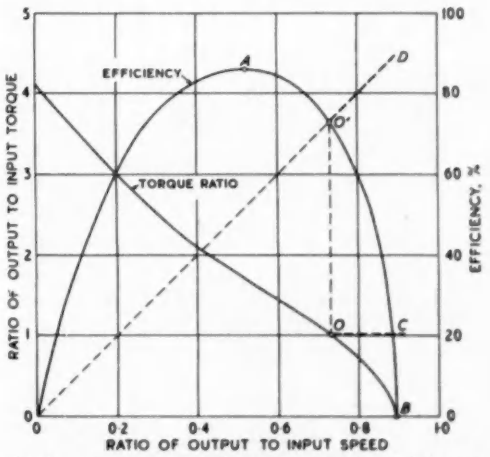


Fig. 65. **Comparison of performance curves of a typical torque converter and a converter-coupling.**

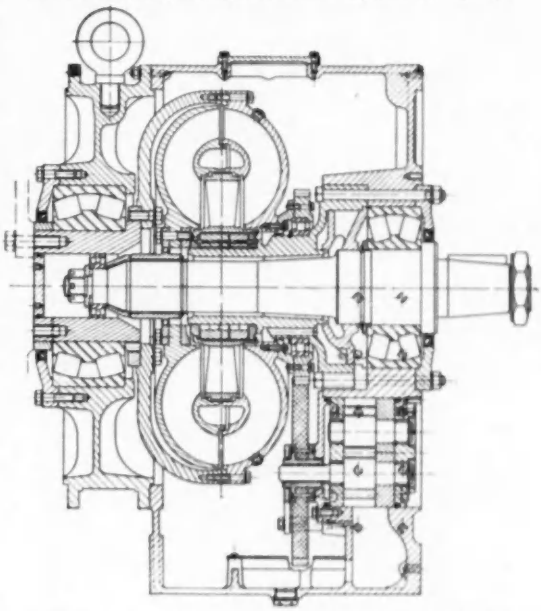


Fig. 66. **Section through a typical Brockhouse industrial converter-coupling.**
(Courtesy of Brockhouse Engineering Ltd.)

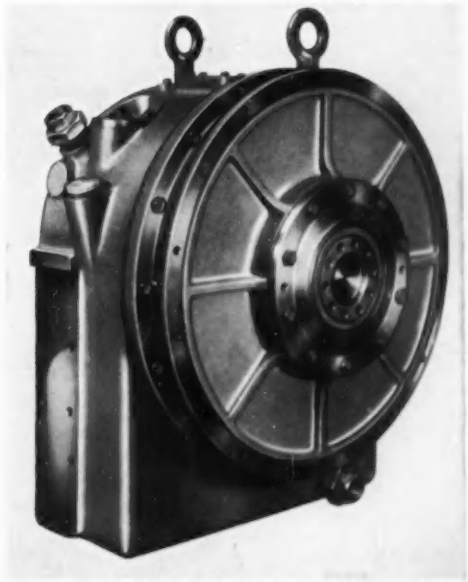


Fig. 67. **View of a typical Brockhouse converter-coupling.**
(Courtesy of Brockhouse Engineering Ltd.)

This condition is overcome in the converter-coupling by mounting the reaction member on a one-way clutch or free-wheel device, so that the reaction blades are free to revolve in the same direction as the impeller and the turbine, but are locked to the frame against the opposite direction of rotation. Thus, the reaction member is enabled to accept torque in the direction necessary for increasing the output torque, but is able to rotate freely

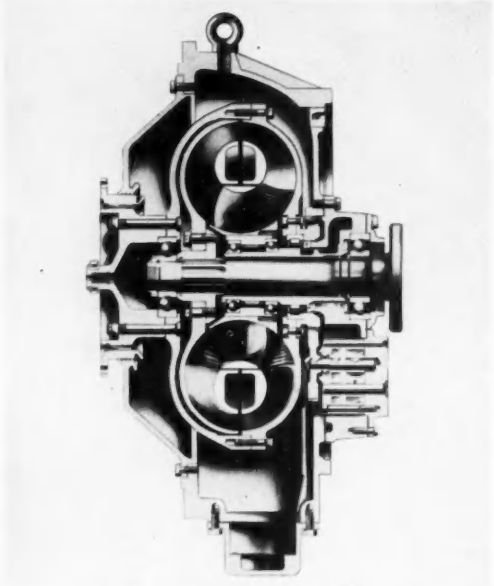


Fig. 68. **Section through a type DF converter-coupling.** (*Courtesy of Self-Changing Gears Ltd.*)

for output speeds above the coupling point. The result is that, as shown in Fig. 65, the curve of torque ratio decreases as far as the coupling point *O* but, instead of continuing downwards to zero along *OB*, it remains constant, taking its subsequent path along the dotted line *OC*. Similarly, the curve of efficiency, after reaching the design point *A*, falls steadily until it reaches coupling point *O'*, after which, instead of dropping to zero along the path *O'B*, it rises again, following the path indicated

Fig. 70. **View of typical type DF converter-coupling, looking towards the input end.** (*Courtesy of Self-Changing Gears Ltd.*)

by the dotted line *O'D*. In other words, when the reaction member is rotating freely, it ceases to be effective and enables the turbine and the impeller to react together as a normal hydraulic coupling, preventing a decrease in torque at high output speeds and producing a sharp rise in efficiency at output speeds above the coupling point.

A cross-section of a typical converter-coupling made in the U.K. by Brockhouse Engineering Limited is shown in Fig. 66. In these units, the turbine and impeller are physically of the same dimensions, so that the two members react to one another with a 1:1 torque in close speed relationship, whether driving or over-running, thus producing a good 1:1 drive at speeds above the range of torque conversion and giving a good degree of engine braking during over-run conditions. The unit is self-contained in a cast-iron housing, with the exception of any particular type of oil cooler which may be required for a given application. The casing also contains a large oil sump and its own circulating pump, filter, and relief valve. Built-in cooling arrangements are available, if required, with all models.

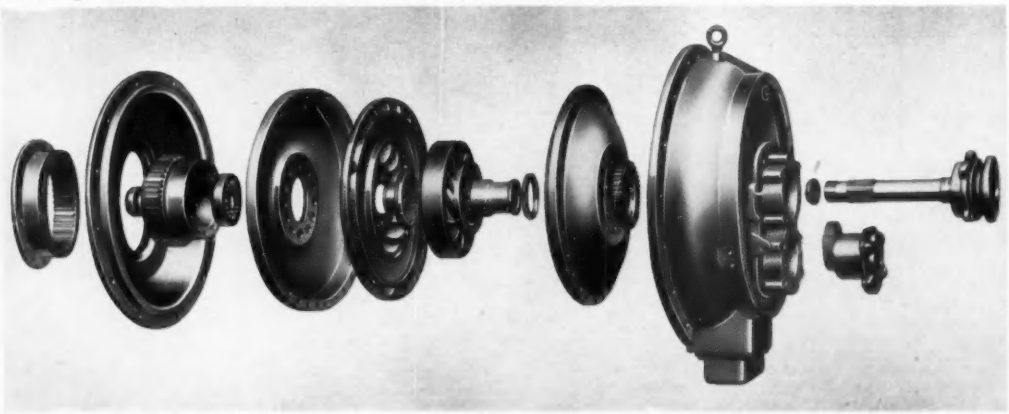


Fig. 69. **Exploded view of the principal components of a type DF converter-coupling.** (*Courtesy of Self-Changing Gears Ltd.*)

These Brockhouse units, which normally provide a torque-multiplication ratio of about 3:1, are available in a wide range of sizes, with capacities from about 200 hp at 2000 rpm up to 900 hp at 1500 rpm. A view of a typical Brockhouse torque converter-coupling is shown in Fig. 67.

A sectional view of a converter-coupling, type DF, manufactured in the U.K. by Self-Changing Gears Ltd., under licence from the Schneider Manufacturing Corporation, U.S.A., is shown in Fig. 68. Available in a wide range of sizes, with wheel diameters from 11 to 19 in., and corresponding maximum power ratings of from about 80 hp at 2800 rpm to 450 hp at 2000 rpm, these simple single-stage units provide a torque-multiplication ratio of 3:1, utilize impellers, turbines, and reaction members made of one-piece aluminium castings, and are kept filled with oil under pressure (30 psi) by an oil pump, which is mounted on the side wall of the converter housing for easy accessibility. These units are also available as pure torque converters (type D), i.e., with a non-rotating reaction member, and (type DFL) as converter-couplings furnished with a direct-drive lock-up clutch, making for added economy when operating in the higher speed range. An exploded view of a type DF converter-coupling is shown in Fig. 69 and a view, looking towards the input end, of an 18-in. type DF converter-coupling in Fig. 70.

An example of a torque converter combined with a hydraulic coupling as separate units is shown in Fig. 71. In this unit, which is manufactured in Germany by J. M. Voith G.m.b.H. and which is basically intended for tractive applications, e.g., in locomotives, the input shaft

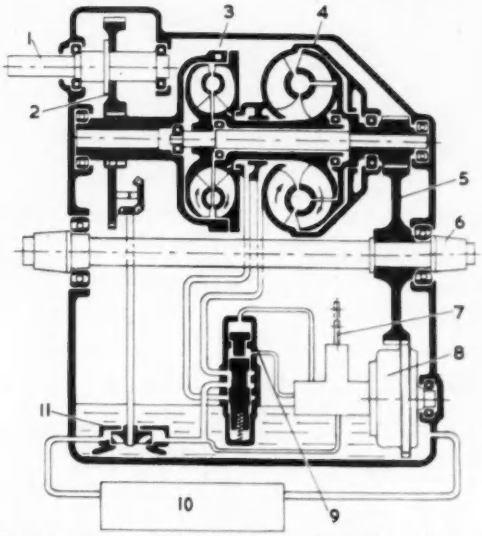


Fig. 71. **Voith transmission with hydraulic coupling and torque converter as separate units.**
(Courtesy of J. M. Voith, G.m.b.H.)

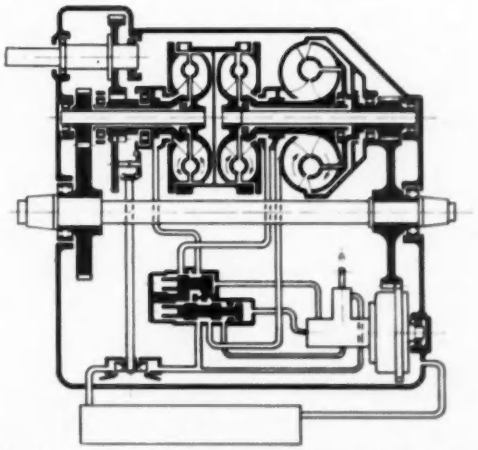


Fig. 72. **Voith transmission with two hydraulic couplings and one torque converter.**
(Courtesy of J. M. Voith G.m.b.H.)

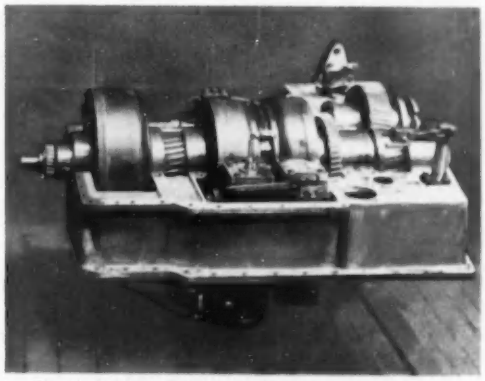


Fig. 73. **Voith transmission with three torque converters.**
(Courtesy of J. M. Voith G.m.b.H.)

(1) drives the hydraulic coupling (3) and the torque converter (4) by step-up gearing (2), this gearing serving to reduce overall dimensions to a minimum by driving the coupling and converter impellers at high speed. In addition, the use of step-up gearing permits, by proper selection of gear ratio, the adaptation of a given transmission size to various prime movers. Similarly, the turbine units are connected via step-down gearing (5) to the output shaft (6).

The working circuits are supplied with oil by a small pump (11) via a piston-type control valve (9) and a cooler (10). Depending on the position of this valve, either both working circuits are empty or one or the other is filled and therefore operative, emptying and filling being accomplished simply and rapidly. Change-over from one working circuit to the other is effected automatically by means of a small governor (8) controlled by the speed of the output shaft (6), the operator being concerned only with switching on or off the unit as a whole. If desired, the spring load on the governor (8) can be varied by connecting the adjustment (7) with the throttle lever of the prime mover, so that, as the power of the engine decreases, the change-over speed also decreases.

In the case of locomotives, when high tractive efforts, e.g., for starting and hauling on steep gradients, and high maximum speeds are necessary, one torque converter may not meet requirements. For this reason, these Voith transmissions, which are available with capacities up to 1400 hp, can be supplied with two or more torque converters, or converters and couplings, combined in one

unit, each circuit being used throughout a certain speed range. For example, a design utilizing two couplings and one torque converter is shown in Fig. 72, and a design using three torque converters in Fig. 73.

In addition, Voith manufacture a comprehensive range of torque converters for industrial applications, including models with normal fixed blades for conventional torque-converter operation, and models with adjustable blades for use with constant-speed motors and providing a variable-speed feature.

Conclusion

In this survey, an attempt has been made to describe the different infinitely variable hydraulic transmissions in use throughout the world to-day, to discuss their characteristics and potential applications, and to detail the products of leading manufacturers.

Although every effort has been made to make it as comprehensive as possible, limitations of space have necessitated a number of omissions. Thus, it is regretted that more space could not have been devoted to infinitely variable drives using fixed-displacement pumps and motors in combination with suitable valving and circuit arrangements, as, despite their comparatively low efficiency at low speeds, the number and variety of circuit arrangements are interesting and demonstrate their flexibility in specific applications. Similarly, no attempt has been made to describe fixed-displacement pumps, including the large number of gear pumps and screw pumps made by a host of manufacturers. This omission is, however, not considered too serious, inasmuch as gear and screw pumps are extremely well known and fixed-displacement pumps of other types, with few exceptions, basically fall into and conform with one or other of the categories described.

It is also true that the names of several makes of variable-delivery pumps and/or motors, including package-type units, have not been given. In a survey of this type, this is inevitable, and such omissions must not be considered as indications that the products concerned are not worth mentioning, when, in fact, the reason is due solely to lack of space.

Finally, it must be stressed once again that the selection of an appropriate hydraulic (or, for that matter, of any) infinitely variable speed transmission depends on a number of factors. Consequently, close co-operation between purchaser and manufacturer is essential, if optimum and economically justifiable results are to be obtained.

Acknowledgments

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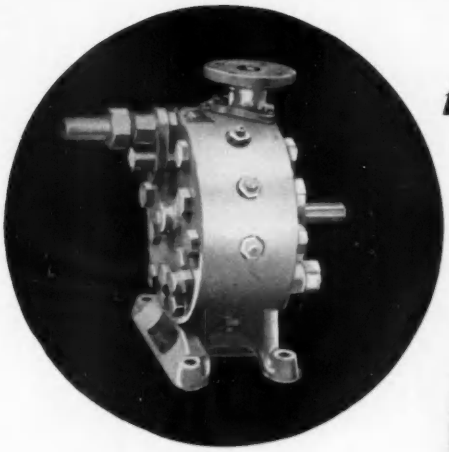

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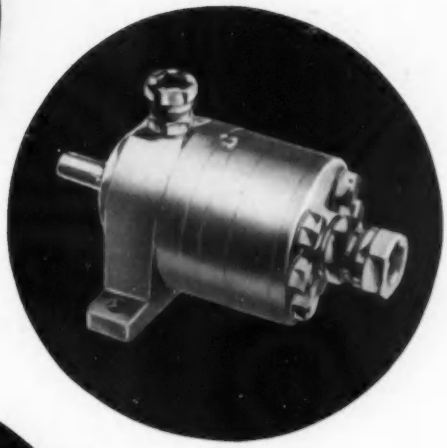

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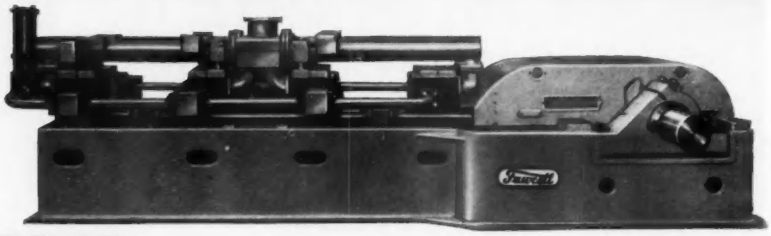

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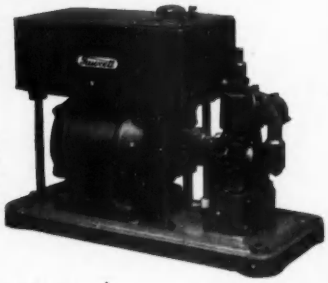

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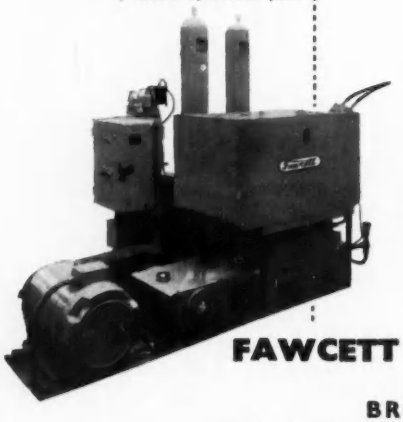

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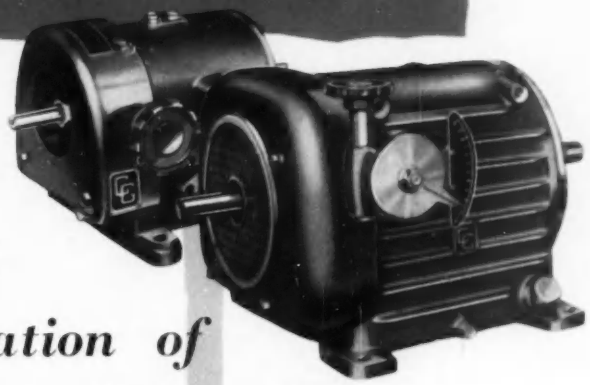

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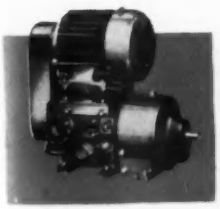

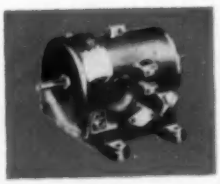
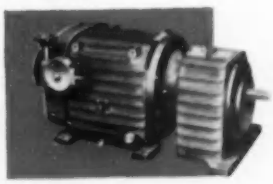

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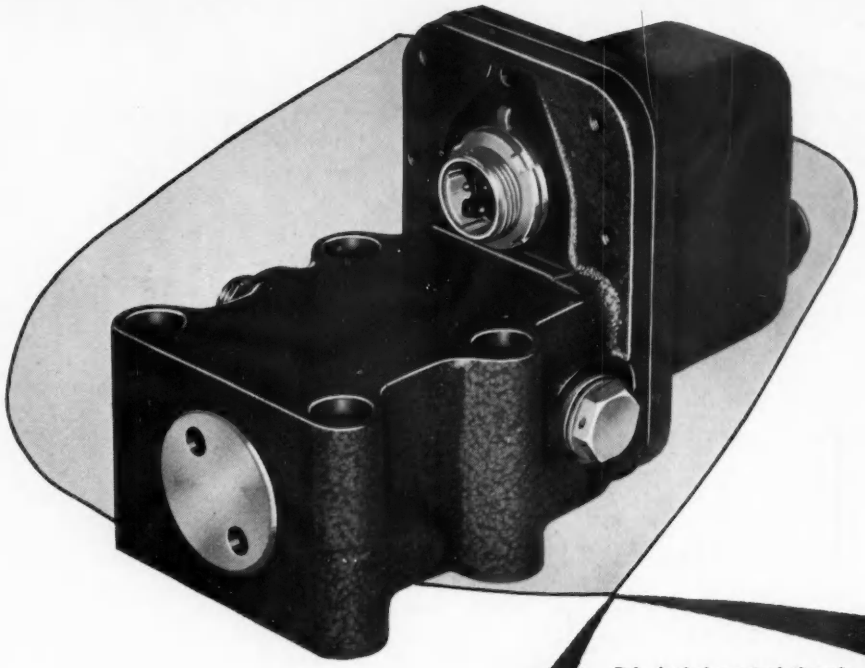

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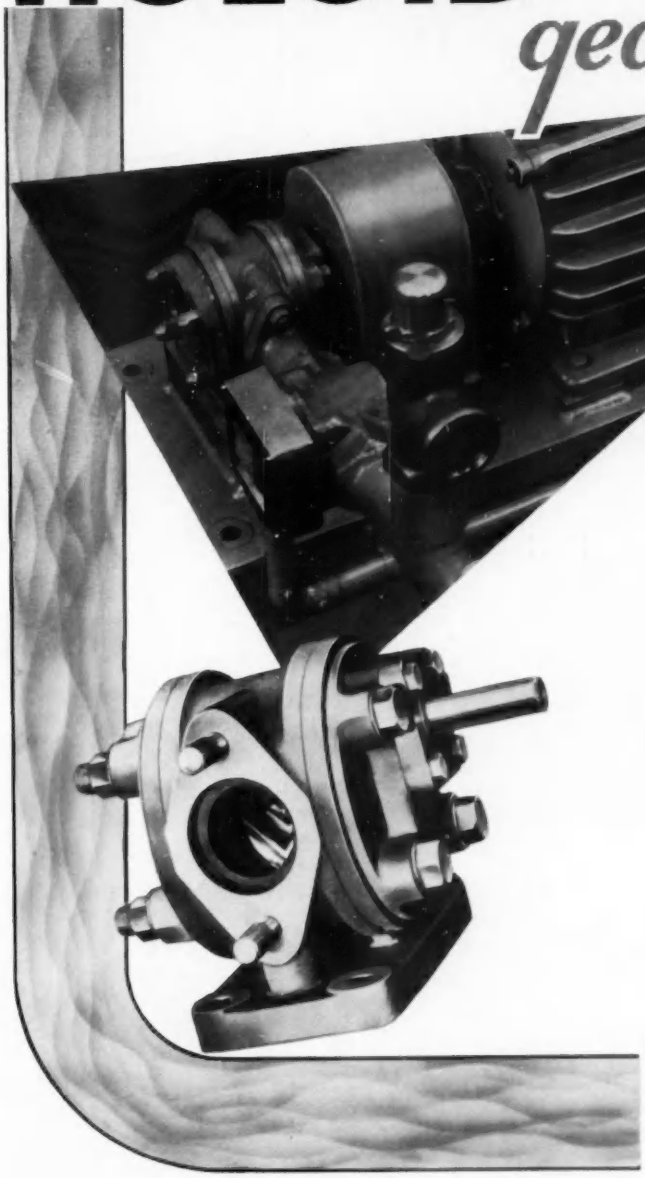

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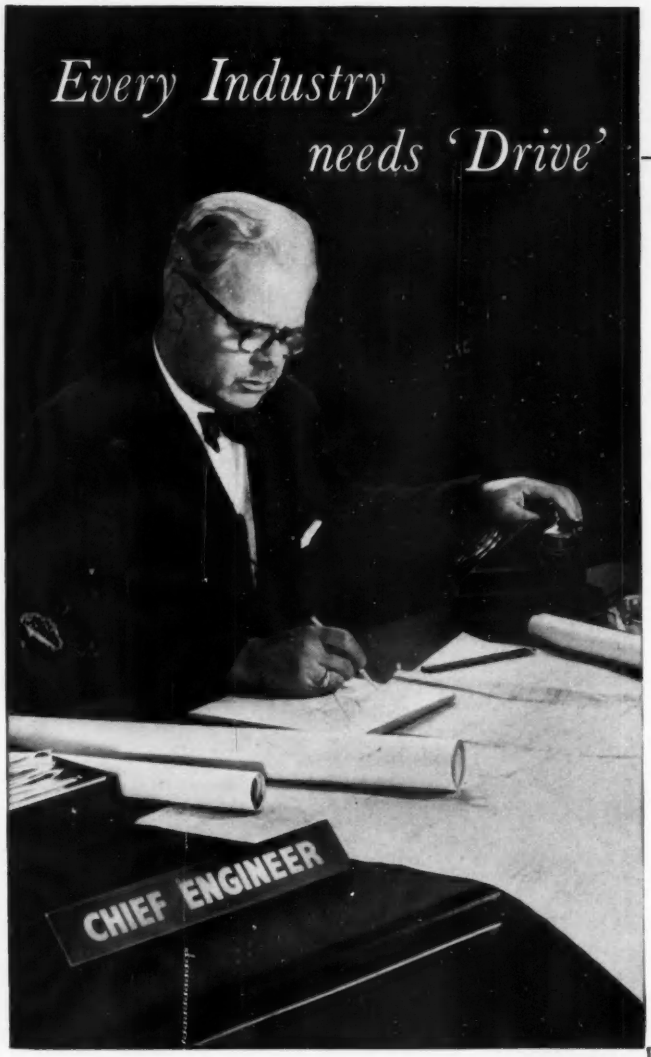

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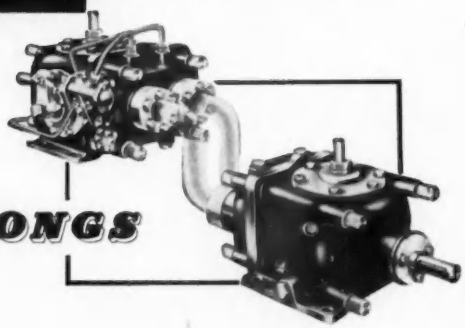

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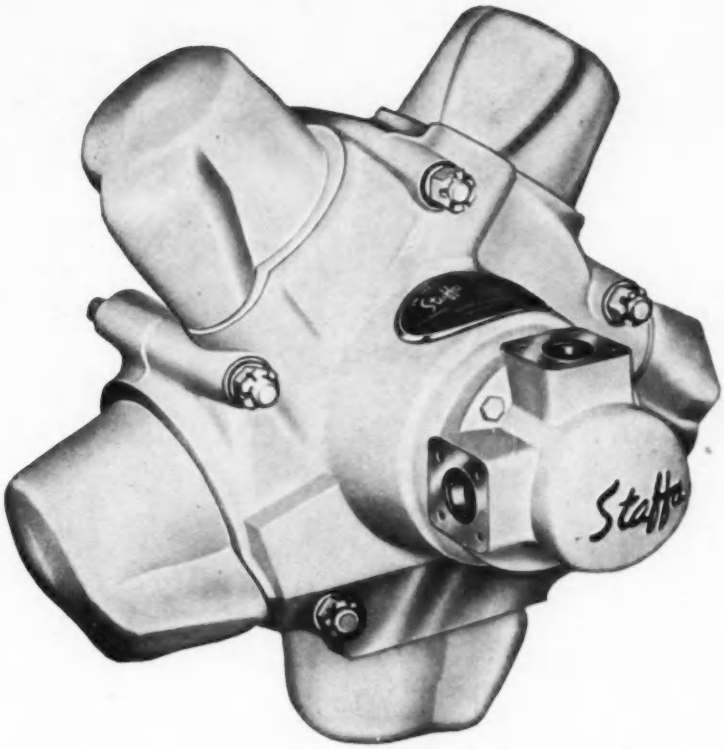

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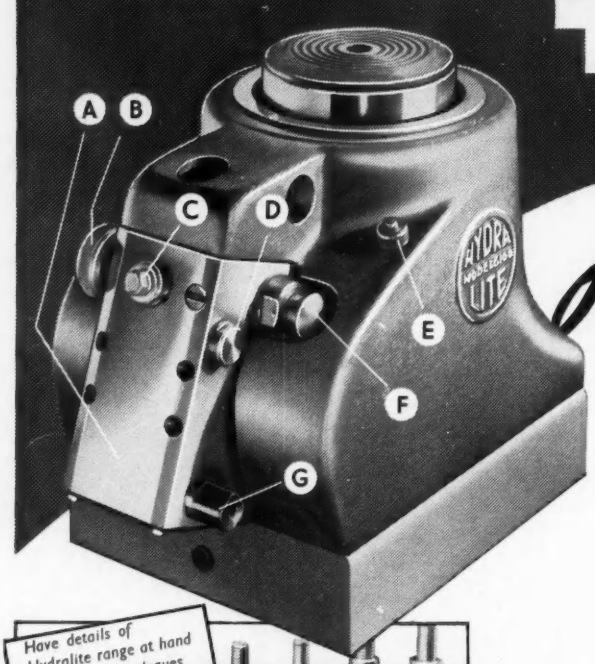

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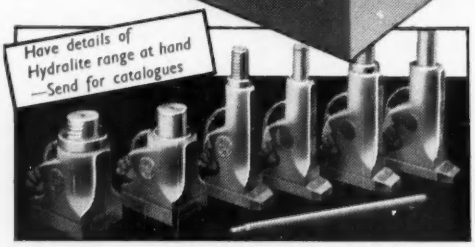

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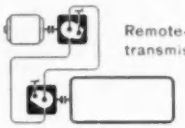

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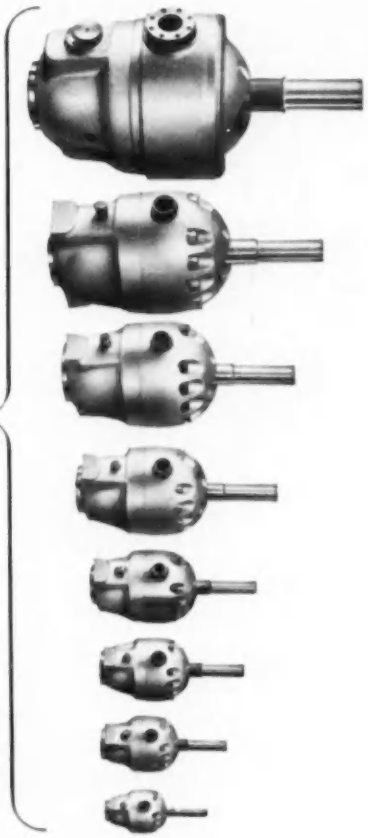

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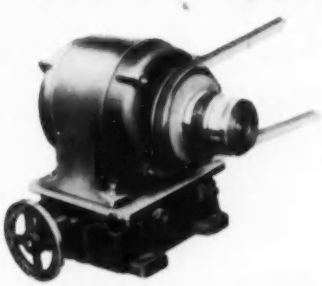

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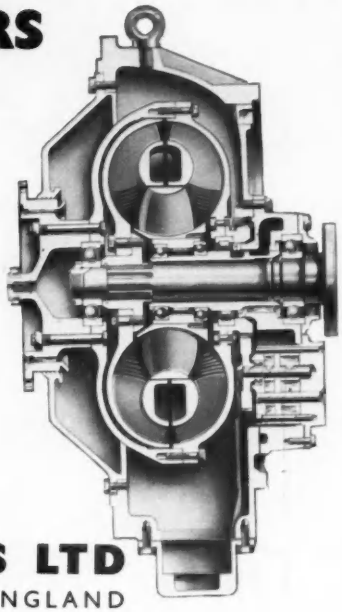

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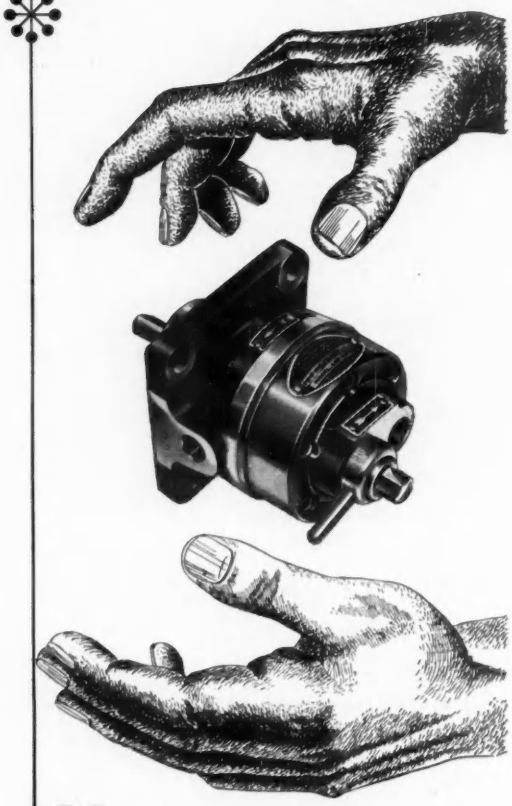

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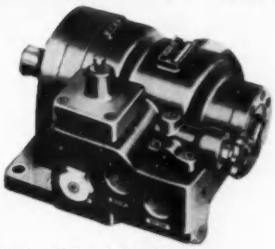

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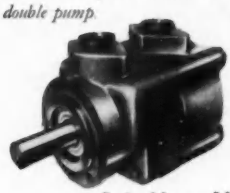

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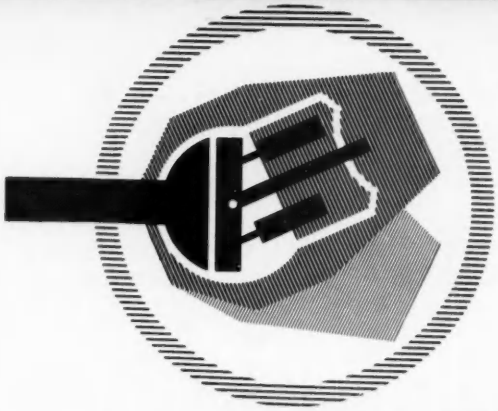

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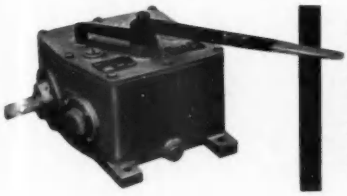

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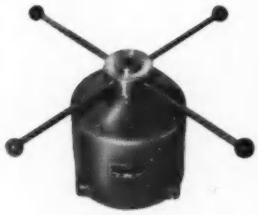

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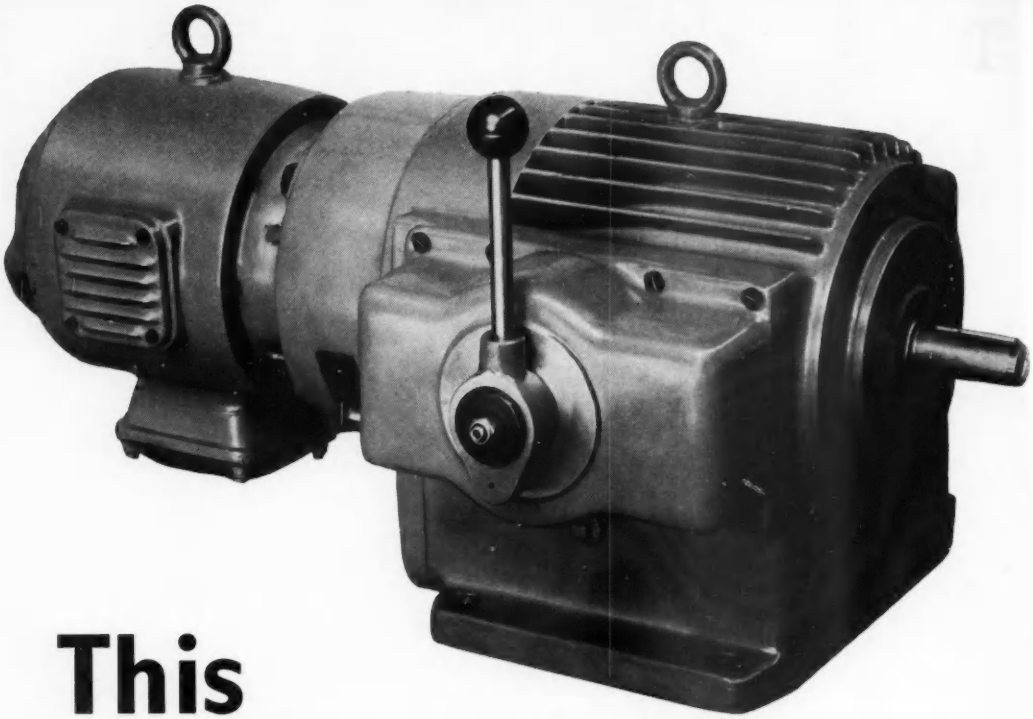

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REVIEW OF WORLD ENGINEERING PROGRESS
VOL. XIX No. 7 JULY, 1958

The Effects of Irradiation on Magnetic Materials

By D. I. GORDON, R. S. SERY, and R. E. FISCHELL. (From *Nucleonics, U.S.A.*, Vol. 16, No. 6, June 1958, pp. 73–77, 9 illustrations.)

CHANGES in magnetic properties induced by reactor radiation are not only important in cases where magnetic devices are required to operate in such environments, but also have implications for solid-state investigations.

The important mechanisms for explaining radiation damage in metals are vacancies, interstitials, thermal spikes, and displacements. Thus, nuclear displacements can be expected to produce the two following macroscopic effects:—

(1) If the solid is previously in equilibrium, this state may be disturbed. Consequently, a non-equilibrium number of vacancies and interstitials will appear, ordered alloys will become disordered, etc. If the temperature is high enough, the damage disappears by annealing, but, if the temperature is low, these effects may not be annealed and permanent damage results.

(2) The material may not be previously in equilibrium but maintained in non-equilibrium by its low temperature. An example is a quenched, disordered alloy which would be ordered if slowly cooled. The irradiation may then actually make the material approach equilibrium.

Since the magnetic properties of ferromagnetic materials are sensitive to changes in atomic lattice structure, neutron irradiation can be expected to affect these properties.

Theory and experiment indicate that displacement effects due to neutron irradiation will be observed in terms of physical property changes, particularly in order-disorder alloys. According to Bozorth[1], 5–79 Molybdenum Permalloy (see Table I), when heat-treated for its optimum magnetic properties, is in a critically ordered state. The marked changes observed in its ferromagnetic properties indicate that the degree of ordering changes significantly.

TABLE I: NOMINAL COMPOSITION OF MAGNETIC MATERIALS TESTED

Material	Percentage Composition
3·5% silicon-iron	3·5 Si, 96·5 Fe
Orthonol	50 Ni, 50 Fe
5–79 Molybdenum Permalloy	5 Mo, 79 Ni, 16 Fe
2 Vanadium Permendur	2 V, 49 Co, 49 Fe
16 Alfenol	16 Al, 84 Fe
Nickel-ferrite	$NiO.Fe_2O_3$
2–81 Molybdenum Permalloy	2 Mo, 81 Ni, 17 Fe

Another order-disorder alloy is 16 Alfenol, exhibiting its optimum magnetic properties in a disordered state[2]. An ordered sample, heat-treated for poor magnetic properties, was irradiated to determine whether it could be improved through disordering by neutron irradiation. The results were positive, i.e., magnetic properties, though by no means perfect, were certainly improved.

It has been reported by Aronin[3] that the intrinsic induction of ordered nickel-manganese alloys (16·5 to 31·9% Mn) shows decreases up to 80% when they are irradiated to about 10^{20} neutrons/cm². For thermally disordered alloys containing 16·5 to 22% Mn, irradiation considerably increases the intrinsic induction. With manganese contents above 23%, the effect is reversed, and the weak magnetic behaviour of alloys in this region is further decreased by irradiation.

Increases of up to 700% in the magnetic susceptibility of some stainless steels have been reported by Leeser[4]. However, these steels are only feebly magnetic before irradiation and, although the fractional increases are large, the total magnetism attained is still relatively insignificant.

EXPERIMENTAL PROCEDURE AND RESULTS

In the present investigation, the changes in magnetic properties of the seven commercially available magnetic materials listed in Table I were measured. For this purpose, measurements were performed on seven toroidal cores specially made with radiation-resistant insulators and mountings. The assembly used permitted remote measurement in the reactor to be made, as well as post-irradiation measurements when the samples were radioactive. Since the organic materials commonly used in toroidal core boxes, magnet-wire insulation, and interlaminar insulating spacers are known to deteriorate appreciably at the neutron-flux level under consideration, it was necessary to use inorganic substitutes. Thus, core boxes were made of anodized aluminium instead of Bakelite, interlaminar insulators were punched from 2-mil sheet mica instead of paper, and glass-covered magnet wire was used instead of Formvar-insulated wire.

The seven assembled cores were mounted in a core-holder and were irradiated in a hole in a reactor for 493 hr. The flux was $1{\cdot}8 \times 10^{12}$ total neutrons/cm²-sec with a cadmium ratio of 1:25, so that 4% of the total neutrons had energies greater than 1 eV. The integrated neutron flux to which the specimens were exposed was $2{\cdot}63 \times 10^{18}$ thermal ($E < 1$ eV) neutrons/cm² and $0{\cdot}11 \times 10^{18}$ fast ($E > 1$ eV) neutrons/cm². The gamma flux was about 4×10^{11} photons/cm²-sec, with a photon energy of about 1 to 2 MeV.

The temperature in the hole varied slowly from as low as 40°C to as high as 70°C, but for the most part remained between 55 and 65°C. To eliminate temperature as a variable, pre-irradiation measurements were made at both hole and room temperatures, and it was found that there was no appreciable difference between the two sets of data obtained. Absence of gamma heating in the samples was indicated by the constancy of saturation during irradiation, as magnetic saturation is

temperature-sensitive, decreasing with increasing temperature.

The magnetic properties measured on all cores before, during, and after irradiation were initial permeability μ (with an induction B of 20 gauss), maximum permeability μ_{max}, residual flux density or remanence B_r, coercive force H_c, and maximum induction B_{max} (with a magnetic field intensity H of 30 oersteds). In addition, d.c. magnetization curves, a.c. hysteresis loops, and magnetization curves at both 60 and 400 cps were obtained. Loss measurements were made on the nickel-ferrite and the powdered 2–81 Molybdenum Permalloy cores before and after irradiation. The magnetostrictive properties of the 16 Alfenol, 2 Vanadium Permendur, and nickel-ferrite cores were also measured before and after irradiation.

Direct-current measurements were made with a standard fluxmeter circuit, while a.c. hysteresis loops were displayed on an oscilloscope, using an electronic integrator in a conventional circuit. For a.c. magnetization curves, the peak field intensity H_p was measured by means of a peak-to-peak voltmeter across a metering resistance ; the peak induction B_p was determined with a rectified, average-reading, vacuum-tube voltmeter.

The core-loss factor was measured as a function of both frequency and flux density. From this information, obtained with a Maxwell bridge, core-loss coefficients were determined.

The magnetostrictive property measured on three of the cores is the electromechanical-coupling coefficient. This is considered to be the most important single parameter in the evaluation of a magnetostrictive material for transducer use.

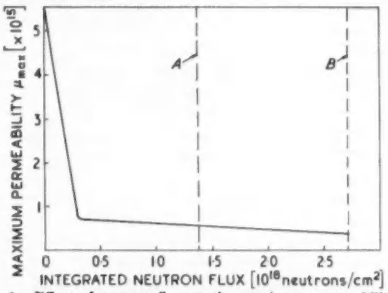

Fig. 1. Effect of neutron flux on the maximum permeability of 5–79 Molybdenum Permalloy.
The dotted lines indicate (*A*) the reactor down for a 15-hr period, and (*B*) the reactor down.

The results of the measurements of the effects of increasing amounts of irradiation on some of the d.c. properties of the samples showed that 5–79 Molybdenum Permalloy and Orthonol suffered the most significant changes, and 16 Alfenol, 3·5% silicon-iron, and 2 Vanadium Permendur the least significant changes. Between these two extreme groups lie nickel-ferrite and powdered 2–81 Molybdenum Permalloy, both of which showed small changes in permeability but substantial changes in core-loss characteristics in the frequency range of 10 to 50 kcps. None of the three cores of 16 Alfenol, 2 Vanadium Permendur, and nickel-ferrite tested for magnetostriction changes showed any appreciable effect.

Permanency of the effects was shown by constancy of peak induction for several materials as a function of time after the end of the irradiation period. Additional measurements of d.c. and a.c. properties at later dates continue to indicate that the changes in properties induced by the irradiation are permanent.

Fig. 1 shows how the maximum permeability of 5–79 Molybdenum Permalloy deteriorates in a reactor environment. There is a drastic drop early in the period, and by the end of the irradiation the maximum permeability has dropped 93%. The other properties are similarly degraded.

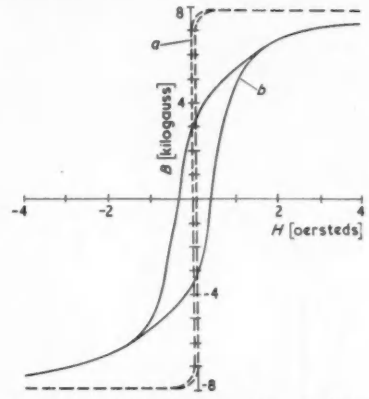

Fig. 2. Hysteresis loop of 5–79 Molybdenum Permalloy (*a*) before, and (*b*) after exposure to 2·74 × 10^{18} neutrons/cm^2.

The effects of increasing amounts of irradiation on the d.c. magnetization curve of 5–79 Molybdenum Permalloy include a saturation effect. Fig. 2, which shows the effects of the total exposure on the 60-cps hysteresis loop, further emphasizes the drastic degradation in the properties of this material. Any device depending on the optimum properties of 5–79 Molybdenum Permalloy would surely fail after exposure to this irradiation.

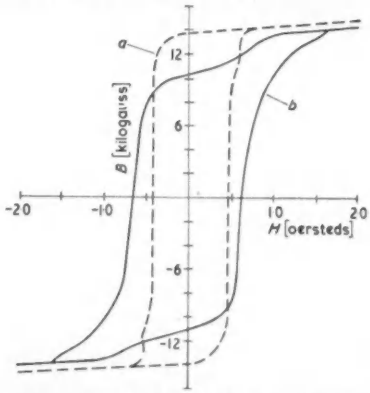

Fig. 3. Hysteresis loop of Orthonol (*a*) before and (*b*) after exposure to 2·74 × 10^{18} neutrons/cm^2.

Orthonol also shows important changes in magnetic properties. The effect on the 60-cps hysteresis is shown in Fig. 3. Decrease in rectangularity, loop distortion, and increase in a.c. coercive force due to irradiation present a problem in the design of magnetic amplifiers.

It may be said that almost all the samples showed deterioration in their magnetic properties. This was, however, to be expected, as they are manufactured for optimum magnetic characteristics. Nevertheless, the magnitudes of effects vary considerably from one material to another.

(*Concluded on page 318*)

Shrink-Fitting by Stress Relief

By A. PEITER. (From *Werkstattstechnik und Maschinenbau*, Vol. 48, No. 5, May 1958, pp. 265–270, 5 illustrations.)

THE distribution of internal stresses in a quenched ring causes it to contract when these stresses are relieved. Consequently, if such a ring is mounted on a shaft with a push fit at room temperature, a shrink fit can be obtained by subsequent stress-relief treatment of the assembly.[1] This two-stage process permits precise seating and controlled shrinkage.

The method evolved by Sachs for the determination of internal stresses in circular cylindrical specimens[2,3] has been adapted by the author for the determination of stress in a shrink-fitted assembly.[4] In these tests, rings of two different steels were used (Tables I and II).

TABLE I : PERCENTAGE COMPOSITION OF STEELS USED

	C	Si	Mn	Ni	Mo	Cu
Ring steel *A*	0·06	0·004	0·32	0·10	0·01	0·08
Ring steel *B*	0·36	0·21	0·67	0·05	0·01	0·09
Shaft steel	0·56	0·22	0·61	0·06	0·23	0·11

TABLE II : PROPERTIES OF STEELS USED

	U.T.S. [tsi]	Elongation [%]	Vickers Hardness [kg/mm²]	Yield Point [tsi] at		
				20°C	300°C	400°C
Ring steel *A*	21	32	84	13·8	7·8	7·4
Ring steel *B*	33	27	136	19·7	13·0	10·7
Shaft steel	44	18	176	24·1	20·8	18·6

In the conventional shrink-fitting of untreated rings on a shaft of harder material, the residual stresses indicate plastic enlargement of the ring and increase with increasing " negative allowance " or interference, as indicated in Fig. 1, which shows residual stresses in the shafts and in two untreated rings of steel *B* after shrink-fitting as a function of both ring diameter and area. In this case, both rings were 30 mm in length, but had different bores and outside diameters, i.e., 30 mm bore and 120 mm o.d. (Figs. 1a and 1b), and 45 mm bore and 90 mm o.d. (Figs. 1c and 1d). The interferences were respectively 0·05% (Figs. 1a and 1c) and 0·25% (Figs. 1b and 1d), and the centre-line average value was between 0·25 and 0·75 μ.

TABLE III : FREE CONTRACTION DURING STRESS-RELIEF TREATMENT OF QUENCHED RINGS OF 30 MM LENGTH, 40 MM BORE, AND 60 MM O.D.

	Percentage Contraction of Bore Diameter after Stress-Relief Treatment at	
	300°C	500°C
Ring steel *A*	0·025	0·05
Ring steel *B*	0·075	0·15

If the ring is hardened by heating and then quenching, compressive residual skin stresses and tensile interior stresses are observed before shrink-fitting (Figs. 2a and 2d). In this connection, the diagrams in Fig. 2, which show residual stresses in the shaft and in quenched rings of steel *A* (Figs. 2a to 2c) and steel *B* (Figs. 2d to 2f), before and after conventional shrink-fitting, refer to rings of 30 mm length and 120 mm o.d., turned to 38 mm bore, then heated and water-quenched, ground to 40 mm bore, and heated in oil before assembly. This heating for shrinkage partly releases these residual stresses and progressively reduces hardness, as shown in Figs. 2b, c, e, and f, which include shrinkage temperatures and Vickers hardness values [kg/mm²].

Stress relief produces geometrical changes, i.e., the quenched ring contracts (Table III) or, if heated, does not expand as much as an untreated ring of identical dimensions. Because of the higher residual quenching stresses in rings of carbon steel, and their consequently greater contraction on stress relief (Table III), this difference in temperature is more pronounced in the case of rings of carbon steels than of mild steels. Thus, for example, for the same interference the ring of steel *B* in Fig. 2 must be heated to from about 100 to 150°C higher than the ring of steel *A*.

With increasing interference, i.e., higher shrinkage temperatures, the radial shrinkage pressure increases, and the final hardness of the ring decreases, as shown in Fig. 2 and also in Fig. 3, in which radial shrinkage pressure between the shaft and rings of steels *A* and *B* is plotted against shrink-fit interference, and refers to rings of 30 mm length, 40 mm bore, and 120 mm o.d., heated for assembly in oil (interference up to 0·25%), or in an electric muffle furnace (0·05% interference). The values of Vickers hardness after shrinking are also shown in Fig. 3. Under practical conditions, however, the values of both radial shrinkage pressure and of final hardness remain above those of the untreated shrunk ring. Obviously, if the shrinkage

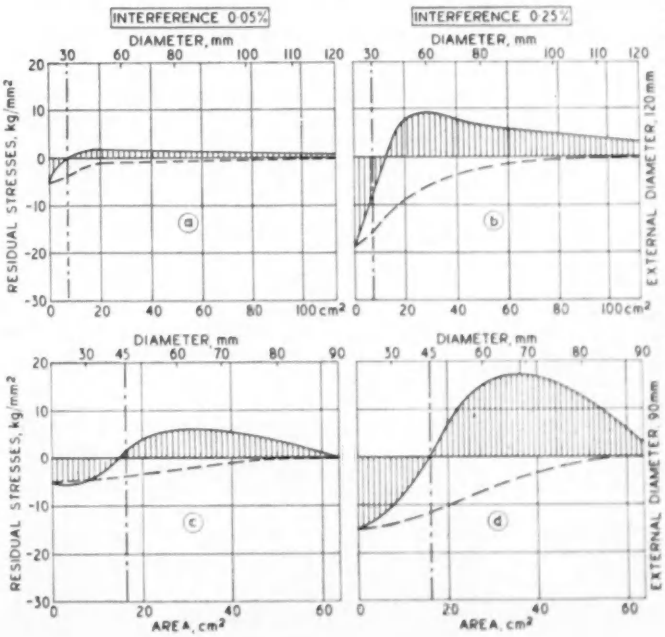

Fig. 1. **Residual stresses in shafts and untreated rings of steel *B* after conventional shrink-fitting.** Solid lines : Tangential stress. Dotted lines : Radial stress.

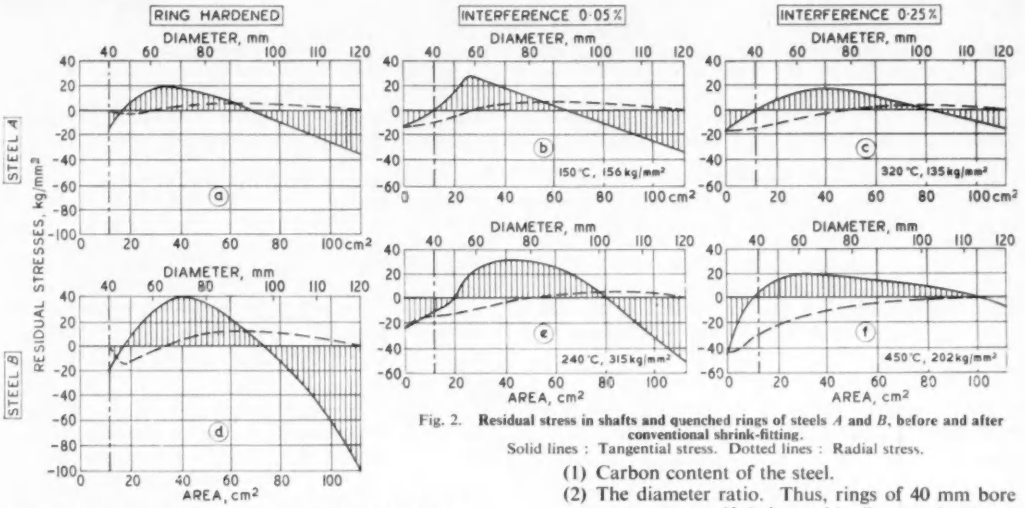

Fig. 2. **Residual stress in shafts and quenched rings of steels *A* and *B*, before and after conventional shrink-fitting.**
Solid lines : Tangential stress. Dotted lines : Radial stress.

temperature is excessive, e.g., above 450°C, the advantages of pretreating are gradually lost, and the values of both pressure and hardness drop towards those of the untreated ring.

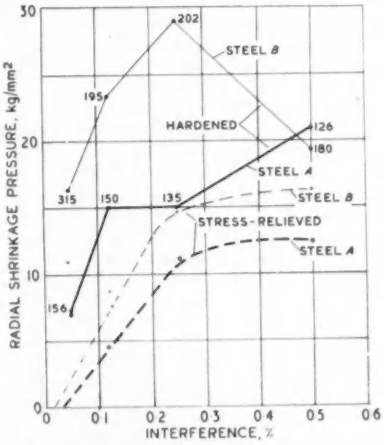

Fig. 3. **Radial shrinkage pressure between shafts and rings of steels *A* and *B* as a function of interference.**

In practice, the ring is machined with a small allowance and is then hardened, after which the bore is ground to finished size and assembled on the shaft with a fit requiring little force for fitting, and the assembly is finally heated for stress relief of the hardened ring. This method has the advantage that the parts can be fitted together carefully at room temperature, without any time restrictions. Alignment of grooves and keyways presents no difficulties, while gears and other parts can be seated and shrunk onto any part of a long shaft without shoulders. Similarly, tubes can be shrunk together accurately, and compound cylinders can be shrink-fitted. A suitable choice of fit, surface roughness, heating temperature, and time permits, to some extent, the combination of a desired shrinkage pressure with a certain residual hardness.

The amount of contraction of the quenched ring during stress-relief treatment depends on the following:—

(1) Carbon content of the steel.
(2) The diameter ratio. Thus, rings of 40 mm bore contract more if their outside diameter is 60 mm than if it is 120 mm.
(3) Stress-relief temperature (Table III), the maximum being reached on complete stress relief at about 650°C.

The effects of (2) and (3) on stress relief during shrink-fitting were investigated systematically. The results are shown in Fig. 5, in which the residual stresses in the shaft and in a quenched ring of steel *B*, before and after shrink-fitting at stress-relief temperatures of 300° and 500°C, are plotted against ring diameter and area for outside diameters of 120 mm (Figs. 5a to 5c), 90 mm (Figs. 5d to 5f), and 60 mm (Figs. 5g to 5i), and in Fig. 4, in which radial shrinkage pressure is plotted against stress-relief temperature for rings of these different outside diameters. For the results shown in Figs. 4 and 5, rings of 30 mm in length and 40 mm bore (ground after water-quenching) were used, and were fitted on the shafts with a clearance of 0·025% or 0·05%, as indicated in Figs. 4 and 5, the centre-line average values being from 0·25 to 0·75 micron. The assemblies were heated to 300° or 500°C in an electric muffle furnace, and cooled in air.

From these results, it will be seen that the shrinkage

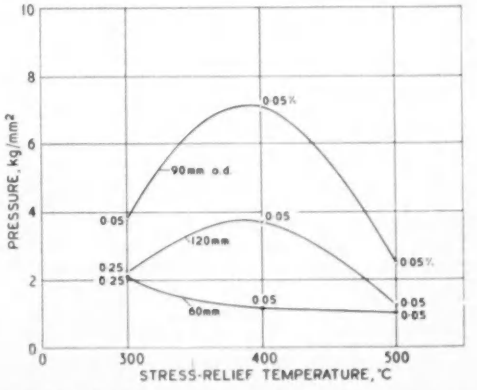

Fig. 4. **Radial shrinkage pressure as a function of stress-relief temperature for assemblies with rings of steel *B* of different diameters, as in Fig. 5.**

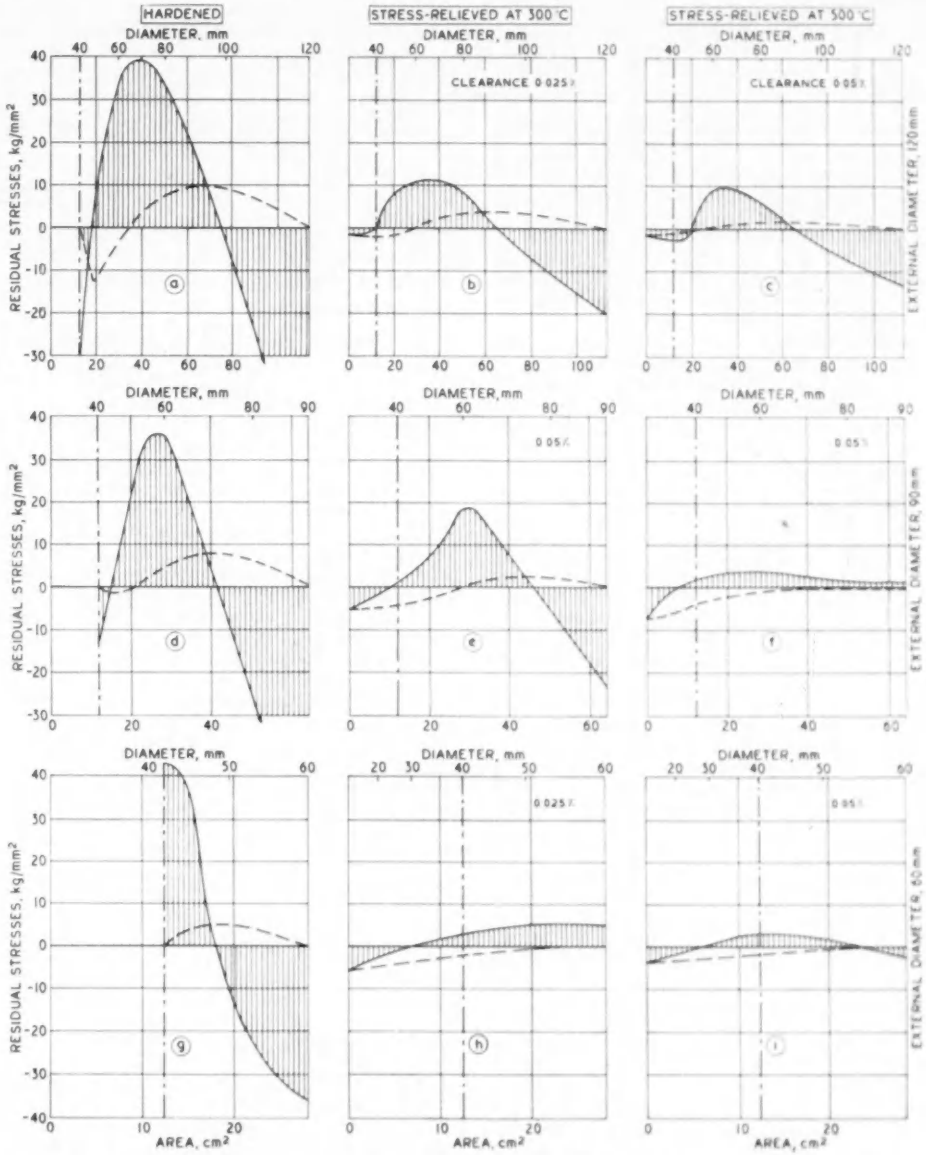

Fig. 5. **Residual stresses in shafts and quenched rings of steel *B*, before and after shrink-fitting by stress relief, for rings of different outside diameters and with different stress-relief temperatures.**
Solid lines : Tangential stress. Dotted lines : Radial stress.

pressure for a ring of 30 mm length and 40 mm bore and 90 mm o.d. is greater than for a similar ring of 120 mm o.d., and both rings show a pronounced maximum shrinkage pressure after stress relief at about 400 °C. This is due to the fact that a higher temperature increases the tendency to contraction and also enables the ring to deform plastically to a greater extent when shrinking onto the shaft, so that its contraction is reduced by plastic flow before the assembly cools down. The ring of 60 mm o.d. showed, after hardening, an exceptional stress distribution, with tension on the inside (Fig. 5), while its shrinkage pressure was lower and had no maximum at 400 °C, as shown in Fig. 4.

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U. S. A.

Reinforced Plastics for Rocket and Missile Applications at Very High Temperatures

By M. W. Riley. (From *Materials in Design Engineering*, Vol. 47, No. 6, June 1958, pp. 100–104, 3 illustrations.)

The problem of selecting materials for service at temperatures ranging from 3000 to 25,000°F and at extremely high gas velocities is at present unique to rocket and missile design. Materials for rocket components may be subjected to two general types of environment, i.e., (1) internal combustion temperatures up to about 5000 or 7000°F and erosion by hot gas streams composed of combustion products, or (2) external skin friction caused by ram air producing in some cases much higher temperatures, possibly up to around 25,000°F.

No structural material known will withstand continuously even the lower internal combustion temperatures, so that the problem is to choose a material that will postpone structural failure for the duration of service. In the case of rockets, the time of exposure to such high temperatures may range from several seconds to several minutes, or in some rare instances (in the lower temperature, range), up to 1 hr. For exposures of such length and at such temperatures, certain reinforced plastics are superior to metals and, in many cases, ceramics. Reinforced plastics pyrolize, but their decomposition rate is usually slower than that of metals. This finite decomposition time can delay structural failure of the material for the relatively brief duration of service encountered in missiles.

The rationalization that seems to explain the low erosion rate of certain reinforced plastics at these high temperatures and high gas flows is rather well developed, although the relative importance of each of the factors involved in the complex process is not established. The three characteristics that seem to be responsible for the high-temperature erosion resistance of reinforced plastics are (1) high heat capacity, (2) capacity to generate surface gas on heating, and (3) low thermal conductivity, as discussed below.

Heat Capacity

Of a variety of structural materials, organic plastics have the highest integrated specific heats from 80° to 7500°F. Since hydrogen has the highest specific heat of any material, the addition of more hydrogen to the hydrocarbon plastic molecule should increase the thermal capacity of the material. Thus, the high heat capacity of hydrogen would seem to explain why organic fibres with high hydrogen contents are such successful reinforcing materials. Reinforcements consisting of nylon fibres have been quite successful, and possibly fibrous materials such as polypropylene or polyethylene, with higher hydrogen contents than nylon, may give even better results. Another approach to providing additional hydrogen content has met with some success, and involves the use of hydrides as filler materials.

A second method of providing higher heat-absorbing capacity during service is to select a material of such composition that, at pyrolysis temperatures, any chemical reaction is endothermic to the maximum degree. For example, when high-silica glass-reinforced phenolic materials are exposed to these temperatures, an endothermic reaction seems to occur.

Gas Generation

Since the heat in an extremely-high-temperature environment is transferred to the material by both radiation and convection, any gases generated by the material should interfere with convective heat transfer and may also act as a radiation barrier.

Of the structural materials, organic plastics generate the most gas per given weight of material at 8500°F under equilibrium conditions. Since hydrogen has the highest gas-generation rate, it appears that increasing the hydrogen content of the material should increase its heat-absorption capacity and also its gas-generation rate.

Thermal Conductivity

Low thermal conductivity in the structural material is desirable to limit temperature effects to the surface area and to retard the progress of decomposition through the material. Limiting the heat penetration also results in higher surface temperatures, possibly providing a closer approach to thermal equilibrium and increasing radiation from the surface back to the environmental gases.

The effects of the lower thermal conductivity of organic-fibre reinforcement can be clearly seen in comparing exposed cross-sectioned specimens of organic-fibre-reinforced and glass-fibre-reinforced materials. The organic-fibre-reinforced specimen generally has a much thinner, more clearly defined charred area. In any case, the relatively low thermal conductivity of reinforced plastics limits the depth of the affected zone, and the unaffected zone has essentially the same properties as the material at room temperature.

Thus, in designing reinforced plastics for such service, section thickness should consist of the thickness required structurally (as determined by room-temperature properties), plus the expendable thickness of materials which will be affected by the heat. Unfortunately, sufficient information is not yet available to permit reliable predictions of the depth of penetration of the heat-affected area for different materials under specific conditions.

Problems in using the Theoretical Approach

The theory of maximum heat absorption and gas generation is inadequate in itself for selection of specific materials, because of the non-equilibrium nature of the process. The surface of the material in actual service rarely, if ever, reaches thermal equilibrium with its environment ; thus, maximum thermal efficiency is not obtained. As the surface of the material melts or vaporizes (depending on the material), the liquid or gaseous products are swept off the surface by the high-velocity gas stream. Of course, the higher the viscosity of the fluids (if the material melts), the more stable is the fluid coating, and the closer the surface may come to thermal equilibrium. Thus, at temperatures near 5000°F, silica-glass reinforcements melt, and the high-viscosity molten silica tends to cling to the surface of the material.

One possible reason for the success of resins such as phenolics is that at high temperatures the decomposition products are simple gases and a carbon residue. The carbon residue may protect the underlying material from the hot gases of the environment, and may also provide a hot reaction zone in which the primary gaseous products are brought closer to thermal equilibrium.

Choice of Suitable Plastics

It is too early to say definitely which reinforced compositions should be used for a given service condition.

Test methods leave much to be desired, releasable data are inadequate, and the objective data available should be considered qualitatively rather than quantitatively.

In general, phenolic resins provide the lowest erosion rates. However, in a group of phenolic materials from various suppliers, there is as much variance in erosion rates as there is in groups composed of different resins, such as silicones, melamines, and epoxy resins.

At temperatures between 12,000 and 22,000°F, phenolic resins with organic reinforcements such as nylon or acrylic fibre seem to provide the best results. In glass-phenolic compositions, the lower the glass content the lower is the erosion rate.

In the lower-temperature range (4000 to 5400°F), phenolics reinforced with high-silica glass fibres or asbestos provide in some cases lower erosion rates than phenolics reinforced with organic fibres. In addition, materials with higher glass content are generally better than those with lower glass content ; evidently, at these temperatures the high-silica glass " sweats " out and coats the surface with silica, increasing the thermal efficiency of the protective mechanism.

With regard to other resin systems, the erosion rates of glass-silicone materials are quite low at relatively low temperatures (3000°F), but the materials themselves have comparatively low structural strength. Some test values have been similar to those for the better phenolics.

Results with epoxy-resin systems have not been too promising, except in one case. However, published test information has not been based on epoxy materials particularly formulated for high-temperature erosion resistance.

Commercially Available Materials

Most of the commercial reinforced plastics publicized as particularly developed for rocket and missile use seem to be intended for the temperature range of 3000 to 10,000°F. Of those commercial materials whose compositions are disclosed, most are phenolic, with either asbestos or silica-glass reinforcement. It should be appreciated that suppliers of such materials are definitely handicapped by security limitations, and no objective test results of proprietary materials are releasable.

Erosion rates for certain asbestos-phenolic proprietary materials are reported to be 0·0052 to 0·0072 in./sec on flat panels exposed to combustion temperatures of 4700°F and gas velocities of Mach 2·5.

One high-silica glass-reinforced phenolic material has been compared by its supplier with asbestos-phenolic materials on a basis of burn-through time at 4500°F with a gas velocity of Mach 0·5, as follows :—High-silica glass-phenolic, 200 sec for a thickness of 0·80 in. ; and asbestos-phenolic, 180 sec for a thickness of 0.85 in. These values are compared with a burn-through time for steel of 80 sec for a thickness of 1 in.

Another major supplier, again limited by security restrictions, claims that a disc of material (resin and reinforcement undisclosed), ½ in. thick, shows " very little erosion and no cracking " after a 20-sec exposure to temperatures ranging from 9000 to 12,000°F.

GERMANY

The Influence of Lead on Engine Oils and its Advantages over Tin as a Protective Coating for Pistons

By H. Kessler and H. König. (From *M.T.Z.*, Vol. 19, No. 5, May 1958, pp. 186–189, 3 illustrations.)

Thin layers of lead or tin, because of their anti-seizing properties, are indicated as protective coatings for the pistons of diesel and petrol engines. After tests carried out some years ago under operating conditions, when the sulphur content of fuels had been increased, it was considered that the advantages of such coatings in reducing mechanical friction were outweighed by their disadvantage of acting as catalysts in some chemical processes occurring in the combustion chambers of certain types of engines. The view then expressed[1] was that diesel-engine pistons should not be lead-coated or tinned. Recent research, however, has shown that this opinion is correct only in certain circumstances.

The results also show that lead, in petrol engines, can have an effect not only via the protective coatings but also when it is used as a constituent in fuels. When chemical reactions take place in the presence of lead, sulphur, superheated steam, and a considerable quantity of oxygen, acids are formed which cause rapid wear of mechanical parts and premature ageing of lubricating oils. Consequently, in petrol engines, it is not significant whether the pistons are lead-coated or not, if the great majority of the fuels employed have a high lead content.

In view of the importance of this subject, it was considered of interest to investigate the effects of lead on engines and lubricating oils more closely than previously, and the corresponding results are given below.

Diesel Engines

From previous investigations it is known that the decomposition of oils is favoured by the sulphurous or sulphuric acid formed during the combustion process, particularly as a result of the increased sulphur content of diesel fuels in recent years[2]. Large numbers of tests yielded results which, at the time, were regarded as being mainly of theoretical or laboratory interest. However, in view of continued demands for lead-coated or tin-coated light-alloy pistons for diesel engines, further tests were carried out, which showed that the catalytic action of the protective coatings, contributing to acid formation and hence to premature ageing of lubricating oils, is not equally pronounced in all engines.

In four-stroke diesel engines the danger is practically nil, while in two-stroke engines the results differ considerably. Thus, it was found that some two-stroke engines are completely immune, and that the greatest likelihood of danger arises in engines with particularly good scavenging, e.g., in supercharged engines.

Thus, the view previously held, i.e., that lead- or tin-coated pistons are basically unsuitable, applies only to particularly well-scavenged two-stroke diesels[3,4], for which the use of leaded pistons is still not advisable. In all other cases, the advantage of a protective coating of lead on the sliding surfaces of the pistons can be utilized without any adverse effects.

The protective layer serves as a means of preventing pitting or scuffing of the piston surfaces in the event of a short-time failure of oil lubrication, particularly during the running-in period. After this period, lubrication in most cases is so reliable that it is not essential for the protective layers to remain intact. In four-stroke diesels, the flow of air enabling oxygen to reach the surfaces of the piston skirt (i.e., the " blow-by " phenomenon during compression) is due to insufficient sealing effectiveness of the piston rings, and occurs mainly in new engines, not yet run-in, and also in engines with extensive wear. In two-stroke diesels, the air can flow over the lateral surfaces of the pistons without having to pass any obstructing seals ; in the subsequent sudden combustion process, sulphuric acid is formed, which causes decomposition of the lubricating oil, and this action is favoured by the fact that the sulphur content of diesel fuels is always greater than that of the fuels for petrol engines.

The laboratory tests, which gave results subsequently confirmed on engines, were carried out with cast test bars of a light-alloy piston material immersed in oil heated to 90°C. To simulate engine conditions, appreciable quantities of air, oxygen, and sulphur dioxide could be introduced into the oil, which also contained traces of moisture. It was found in these tests that oil decomposition was much greater as a result of contact with lead-coated test bars than with uncoated bars immersed during the same length of time.

The bars were machined as in the normal process of piston manufacture, so as to obtain a smooth surface. The uncoated bars, after removal from the bath, were covered with an oil film and had only a very small quantity of soft, sludge-like residue. The lead-coated bars, which had a soft, velvety surface before immersion, exhibited at the end of the tests a very rough surface with crevices between the agglomerations of oil and carbon residues. The amount of residue on the coated bars was about 50 times greater than on the uncoated bars, and in engines would have led to pitting. As, in supercharged diesels, combustion temperatures are considerably higher than in petrol engines, although the ignition frequency is lower, the acid formation due to catalytic action of lead when the piston skirts are accessible to oxygen and sulphur is always considerably greater.

Petrol Engines

As previously stated, reaction temperatures are generally lower in petrol engines than in diesel engines. The acid formation due to the catalytic action of lead reaches only small proportions, so that there are no reservations regarding the use of lead- or tin-coated pistons for petrol engines. The presence of lead on piston skirts is insignificant when lead reaches the combustion space more effectively, i.e., though the fuel itself. This finding has been confirmed both in laboratory and in field tests.

The laboratory tests were carried out to determine the causes of sludge formation and its dependence on the lead occurring (a) in tetra-ethyl-lead fuels, and (b) on lead-coated piston surfaces. Only good-quality proprietary lubricating oils were used for the tests, which were carried out in the presence of moisture and after introducing sulphur dioxide and oxygen into the oils.

It was found that there was noticeable sludge formation when lead-coated specimens were immersed in oil, whereas an equally noticeable, but much smaller, quantity of sludge occurred in the oil samples in contact with the uncoated specimens. Consequently, the fact that lead has some influence could not be easily dismissed, but whether it was of sufficient importance to affect engine performance was an open question, since the maintenance schedules for engines always call for a change of oil after specified periods, to eliminate sludge and to ensure proper operating conditions. Consequently, tests were also effected in this connection. The results showed that, in the presence of sulphur and oxygen, the decomposition of oil is much greater with lead introduced through fuels having a high tetra-ethyl-lead content than with lead occurring on lead-coated pistons used in conjunction with undoped fuels.

Advantages of Lead over Tin for Piston Coatings

There are definite advantages in using lead, instead of tin, as a material for thin protective layers on pistons. The normal operating temperature of pistons is generally below 300°C, but higher temperatures can occur, either at localized positions or over large areas of the skirt, if there is a tendency towards scuffing, due, for instance, to an insufficient supply of oil. Therefore, a metal used as a running-in layer or as a protective layer will be particularly successful if it can sustain a higher range of temperatures. Lead has a melting point of 327°C, compared with 232°C for tin, and it is evident that, for otherwise comparable sliding properties, the advantage is with the metal having the higher melting point. Tin, applied as a thin layer on a piston surface, contracts into small spherical droplets when it is heated above its melting point, owing to surface tension. These droplets are cast off or scraped off during piston travel, so that, after an extremely short period no layer of tin is left, particularly on piston-skirt positions exposed to considerable danger of pitting because of high temperatures and pressures.

Because of the good thermal conductivity of aluminium-alloy pistons, local temperatures above 327°C will occur much more rarely than temperatures of 232°C ; consequently, lead offers a better and more reliable protection against piston scuffing or pitting than tin.

The demand for tin coatings is by no means infrequent, and is generally based on their visually attractive bright appearance, as compared with lead coatings, which are dark-brown. It would appear, however, that this is the only point in favour of tin coatings, inasmuch as the superiority of lead-coated pistons seems to have been definitely established. Thus, for example, in tests carried out on a tin-coated piston heated to 250°C for about 15 min, the tin agglomerated into spherical droplets, as already mentioned, and these could easily be removed by slight shaking or scraping, while the remaining portions of the tin layer formed loose patches that were ultimately thrown off the surfaces of the piston skirt, which was at a higher temperature than the melting point of tin. On the other hand, tests on a lead-coated piston established that, after exposure to a temperature of 310°C for 15 min, the condition of the lead coatings on the surfaces of the skirt remained unchanged.

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Properties and Applications of Epoxy Foam Materials

By D. D. Scott. (From *Machine Design*, Vol. 30, No. 12, June 12, 1958, pp. 127–129, 3 illustrations.)

Epoxy foam materials combine the desirable characteristics of cured epoxy resin with the physical properties inherent in hardened foam materials with non-interconnected-cells. They have excellent chemical resistance and dielectric properties, toughness, good adhesive qualities, and a degree of heat resistance which is superior to that of all thermoplastic foams.

The heat-distortion point of epoxy foams is generally superior to those of other thermosetting foams. Of the wide variety of fillers available for strengthening epoxy foam, the one most suitable for a particular application must be selected by testing individual formulations.

Sheets and blocks of epoxy foam must be oven-cured under controlled conditions to realize fully the high-temperature qualities of epoxy materials. In-place curing provides excellent adhesion of the foam to most metals and materials and is practical if parts can be placed in ovens for prolonged cures. If in-place curing is not feasible, performed foams may be readily cut, sawed, drilled, and machined to size before cementing in place. In this connection, epoxy foams, because of their fine, uniform structure and lack of thermoplasticity, have excellent machinability and do not soften under the heat of cutting. Undercuts, angles, and complex contours are easily shaped, and tolerances are comparable with those attained in woodworking.

Potting or encapsulating small components is feasible on a production basis with epoxy foams. To date, successful potting of components up to 3 cu in. in volume has been achieved. The most practical foam density for potting is from 4 to 9 lb/cu ft. Mould design considerations include the use of a suitable mould-release agent to facilitate removal of parts.

Successful potting depends to some extent on the configuration of the part. A very thin section might prove difficult to cast, although location of the thin section relative to the mould must be considered. During the early development of these materials, dissipation of exothermic heat was a problem, owing to their excellent insulating properties. Now, however, closely controlled conditions during curing have eliminated this problem, and epoxy-foam blocks, sheets, and slabs of large cross-section can be successfully produced, with excellent cell uniformity and no charring.

Epoxy foams have exceptional dimensional stability. Foamed pieces display complete freedom from warping and excessive dimensional changes on exposure to temperatures up to 240°F and to high humidity. General-purpose epoxy foams have a heat-distortion temperature of approximately 200°F, but special foams are available which do not soften appreciably at temperatures up to 240°F. Current experiments indicate that this limit will soon be raised. In addition, it is interesting to note that a special self-extinguishing epoxy foam with excellent high-temperature resistance has been produced. Thus, even after several minutes' exposure to the flame of an acetylene torch, surface charring occurred, but the foam did not burn.

Epoxy foams provide an ideal potting compound for electronic components and sub-assemblies, if provision is made for heat dissipation other than through the foam. Substantial weight savings can frequently be made by potting with epoxy foam. Completed foam-potted units are still capable of meeting the strength and shock requirements imposed on solid-epoxy encapsulations.

Foams with densities ranging from 2 to 20 lb/cu ft are formed and cured in place around electronic circuitry and components inside a plastic or metal case. The electrical properties of the foam package are similar to those achieved with solid-epoxy encapsulated units. Replacement of components is accomplished more readily with foam-packaged units than with solid-plastic potted units.

Insofar as water absorption is concerned, the completely unicellular structure of epoxy foams confines moisture pick-up almost exclusively to the exposed outer cells. As an example, a sample of foam with a density of 3 lb/cu ft, after being submerged in 8 ft of water for 48 hr, was found to have absorbed only 1·5 times its weight of water. Epoxy foams can withstand hydrostatic pressures of 500 lb/sq ft, and the degree of water absorption depends on whether the foam structure has a protective outer skin or an exposed cellular layer.

With regard to their chemical properties, epoxy foams are non-corrosive to metal surroundings, and inserts are highly resistant to oils and greases and can be coated with cold-temperature polyester or epoxy skins without pretreatment. No apparent significant physical or chemical changes have been noted in samples tested after a service period of one year.

The unicellular structure of epoxy foams is responsible for their excellent thermal-insulation properties and buoyancy. However, because of this cell structure, they do not qualify as satisfactory acoustic materials, as compared with polyurethanes and phenolics, which are examples of foams having an interconnected cell structure and which do not have the thermal properties or buoyancy of epoxy foams, but have superior acoustic properties.

Thermoplastic polystyrene foams, which are similar in structure to epoxy foams, have the same good properties of thermal insulation and buoyancy. However, polystyrene softens when subjected to slightly elevated temperatures and is nearly liquid at 160° to 180°F. Polyurethane, phenolic, and epoxy foams are all thermosetting and withstand elevated temperatures. At low temperatures, both epoxy and polystyrene foams have superior electrical insulating and dielectric properties. At high temperatures, epoxy foams are superior to the phenolics and polyurethanes, because of residuals remaining in the latter two foams.

An outstanding feature of epoxy foams is that they can be coated directly with polyesters without pretreatment of any kind. Styrene foams, however, must be coated thoroughly with an epoxy to prevent the styrene in the polyester from dissolving the foam. Polyurethanes can be coated with polyesters, since they are not attacked by the styrene. Some phenolics can be coated easily, if the acid residual is not too great.

Because of their light weight, rigidity, insulating properties, and ease of preparation, epoxy foams are generally suitable for applications where structural members with high strength-to-weight ratios are needed. Epoxy foams are also used as filler materials to which protective skins are glued, and for thermal and acoustic insulation. Potential uses include cores for radomes, fuel-tank baffles, insulation panels for refrigerators, and cores for

(*Concluded on page 314*)

Protective Systems for Alternators

By W. BLENDINGER. (From *The Brown Boveri Review*, Vol. 45, No. 2, February 1958, pp. 59–64, 6 illustrations.)

FOR the protection of alternators, various systems can be employed. Thus, a system of protection against asymmetry prevents damage to the alternator when the three phases are unevenly loaded, while supervisory systems cater for the possibility that a sudden interruption of excitation current may have harmful effects. Similarly, in the event of insulation breakdown, it is desirable to incorporate a system of field suppression to reduce or eliminate the magnetic field very quickly.

PROTECTION AGAINST ASYMMETRY

Asymmetric loading may be harmful in three-phase synchronous alternators, because additional losses occur in the rotor, which may consequently become inadmissibly hot. This applies particularly to turbo-alternators having no damper winding, where, for instance, a difference of more than 10% of the rated current between the current in one phase and in the two others may become dangerous, if the condition persists for an appreciable time ; in fact, the additional rotor losses vary with the degree of asymmetry. Alternators with damper windings permit greater asymmetric loading, a factor which can be allowed for when dimensioning the damper winding.

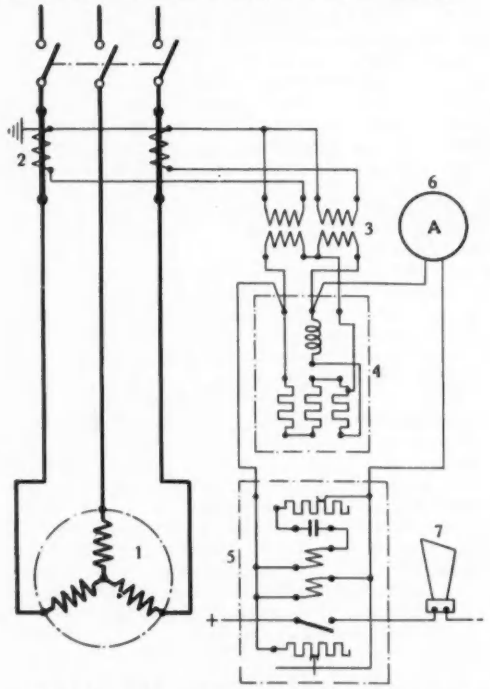

Fig. 1. System of protection of a three-phase alternator against asymmetric loading.

In turbo-alternators without a damper winding, the admissible degree of sustained asymmetry is approximately 8%. When two phases carry their full normal current and the third 90%, the negative-sequence current amounts to 6·6% of the rated value. Maximum asymmetry is produced when one phase is interrupted. If, in these circumstances, the other two phases were to carry the full rated current, this would result in a maximum asymmetry of 57·7%.

A protective system employed by Brown Boveri is illustrated in Fig. 1 and basically comprises a sensitive reverse-current relay (5), a filter (4), and a current-indicating instrument (6), which is fed through a pair of auxiliary instrument transformers (3) connected to the current transformers (2) of two phases from the alternator (1). The filter (4) contains two equal impedances, one being purely resistive and the other lagging by exactly 60 deg. The positive-sequence components of the two primary currents, which are equal but mutually displaced by 120 deg., thus produce equal voltage drops in these impedances, in the same direction, so that at the output terminals there is no voltage difference to feed the relay. The negative-sequence components, however, produce voltage drops displaced by 120 deg., which may be added vectorially at the output terminals by multiplying by $\sqrt{3}$. The relay connected to the output terminals thus experiences a voltage proportional to the negative-sequence component. This relay is normally adjustable between 7 and 15% asymmetry and actuates an alarm (7).

If desired, a two-stage arrangement can be employed, using a second relay (5), which can be set for asymmetry between 20 and 40% and which disconnects through a time-lag device.

The current-indicating instrument (6), connected in series with the reverse-current relay (5), is also calibrated in terms of percentage of rated current and provides a constant record of asymmetry, providing a useful means of supervision.

SUPERVISION OF THE EXCITATION CIRCUIT

Another disturbance which can have harmful effects is the inadvertent interruption of the d.c. excitation of a synchronous machine ; this might be caused by tripping of the field switch, by a connection breaking loose, or by the breakage of a conductor in the excitation circuit.

If the alternator operates alone, feeding its own system, the voltage will collapse and the supply of current will cease. However, if the machine runs in parallel with others or with a power system, after interruption it draws the power necessary for excitation as reactive power from the system and continues at first to run under-excited. The reactive power and magnetization current which the machine previously supplied must now be made good by the other machines, and in certain circumstances this can lead to the latter becoming overloaded, particularly when only a few units are concerned with the interconnection. There is then a risk that the overload protection of these machines will come into action, shutting them off and finally causing the entire system to collapse.

The effect on the machine is not always the same. Turbo-alternators, in general, fall out of step fairly soon and continue to run at a supersynchronous speed as asynchronous generators, without the occurrence of harmful load-hunting. In contrast, salient-pole machines, even at zero excitation, possess a synchronizing torque which, however, can only prevent the machine from falling out of step at very low active loads. If such a machine falls out of step, heavy load-hunting at the slip frequency can commence and this can seriously disturb the parallel operation. The affected machine must therefore be disconnected from the system without delay. Hence, protective devices responding to interruption of

the d.c. excitation are being increasingly employed. A supervisory device of this nature is primarily used by Brown Boveri to protect large turbo- and hydro-electric alternators. Depending on the operating conditions, one of two different versions may be employed, as follows :—

(1) A simple type for use with alternators which, when connected to their system, never run with less than half their no-load excitation. This usually applies to alternators with ordinary self-excited exciters having no pilot exciters. This protective device principally comprises a maximum-resistance relay to keep a check on the rotor circuit, and an under-voltage relay for the exciter circuit. An additional time-lag element prevents the protection from tripping unnecessarily when the excitation current drops temporarily, as can easily occur when the alternator sheds load suddenly.

(2) A type for use with alternators in which the excitation current in the rotor under service conditions may be made less than half the no-load excitation, being, in certain circumstances, zero or even negative. Such alternators are usually equipped with main and pilot exciters, as shown in Fig. 2. The particular running conditions are encountered when an alternator has to operate with a heavy leading load, above all when feeding long transmission lines or an extensive underground network at no-load or with only a very light active load.

The protective device must not only supervise the rotor current alone, but also the field current of the main exciter and the voltages of all excitation circuits. The numerous phenomena, all quite distinct from each other, which can lead to interruption of excitation necessitate using a corresponding number of protective elements.

The supervisory device shown in Fig. 2 includes two maximum-resistance relays (5a and 5b) for supervising respectively the main excitation circuit (rotor) and the excitation circuit of the main exciter, an under-voltage relay (5c) for keeping a check on the voltage of the pilot exciter, and a time-lag relay (6) which, after an adjustable delay, controls the complete de-excitation of the alternator and its disconnection from the system.

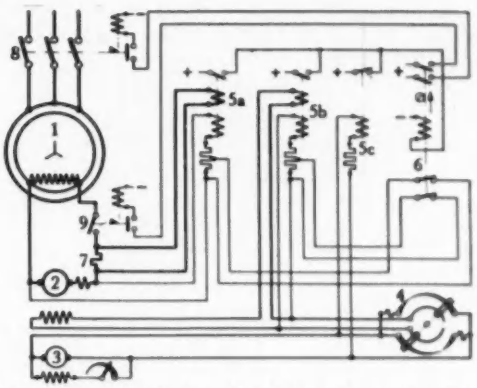

Fig. 2. System of protecting against inadvertent interruption of the excitation circuit of a synchronous alternator with main and pilot exciters.

(1) Alternator.
(2) Main exciter.
(3) Pilot exciter.
(4) Automatic voltage regulator.
(5a, 5b) Maximum-resistance relays.
(5c) Under-voltage relay.
(6) Time-lag relay.
(7) Shunt.
(8) Alternator breaker.
(9) Field switch.

Rapid Field Suppression

Under normal operating conditions, it is desirable, particularly with self-excited machines, for a small amount of residual magnetism to remain in existence, even when the machine has stopped altogether. This remanence ensures that the machine is excited in the correct direction when it starts up again. This is also true of the pilot exciters of large alternators, whose main exciters are separately excited by pilot exciters. In contrast, provided that there is no internal defect in the machine, it is unimportant to the main exciter and the alternator itself whether disconnection of the excitation causes the magnetic field to disappear partially or completely. The field is always built up again by the pilot exciter.

Conversely, in the event of an insulation breakdown in the stator winding of the alternator, it is very desirable for the magnetic field to be suppressed as rapidly and as completely as possible, in view of the damage which can be caused by an arc to the copper, the iron stampings, and the insulation.

In the simplest and cheapest known method of field suppression, the field winding of the exciter is disconnected from its source of supply and short-circuited across a resistance. This arrangement reduces the field relatively slowly and is therefore suitable for use with small units only. It is only a short step to apply the same principle to the field of the alternator itself (Fig. 3). This increases the rate of field suppression in the alternator quite appreciably. In both cases the energy in the magnetic field is dissipated as heat in a resistance, after the field switch is tripped.

The general increase in unit ratings of machines in recent years has been accompanied by a steady trend towards what is called rapid field suppression. This applies more or less to all methods in which the process of de-excitation is initiated by reversing the polarity of the exciter, causing it to run temporarily as a motor. The energy in the magnetic field is thus no longer converted only into heat but also into mechanical power.

The system of rapid suppression employed by Brown Boveri is illustrated in Fig. 4, in which the contacts are shown in their normal service position. If, owing to a

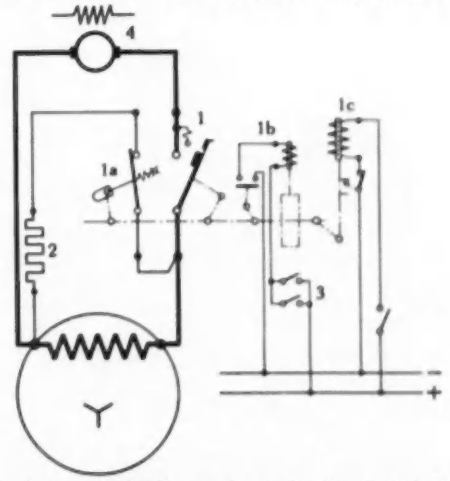

Fig. 3. Conventional field-suppression circuit with main field switch for medium and large alternators.

(1) Main contact.
(1a) Short-circuit contact.
(1b) Tripping solenoid.
(1c) Closing solenoid.
(2) Field discharge resistance.
(3) Contacts of protective relay.
(4) Exciter.

relay picking up or by deliberate operation, the field switch (4) is opened, the exciter voltage is reversed with the aid of auxiliary contacts and made to bypass the voltage regulator (7). The field winding of the main exciter then carries the full voltage of the pilot exciter but with reversed polarity. The higher the voltage applied for counter-excitation the more rapid will be suppression. The over-voltages experienced by the field winding are mostly within tolerable limits, without special precautions having to be taken. The rotor current tends to rise to a negative value, limited by the resistance (6). Owing to the resistance connected in series with the rotor winding, the time constant of the excitation circuit is considerably diminished, so that the current changes very rapidly. The instantaneous under-voltage relay (8), which is fed from auxiliary contacts on the field switch, supervises the alternator voltage and disconnects the exciter contactor (5) just before the voltage reaches zero.

If, for any reason, the under-voltage relay does not trip, a time-lag relay (9) comes into action. Its operating time is so adjusted that it disconnects the contactor (5) as soon as the period normally required for the excitation to drop to zero has elapsed. The resistance (6a) prevents inadmissible overvoltages when the negative excitation is interrupted.

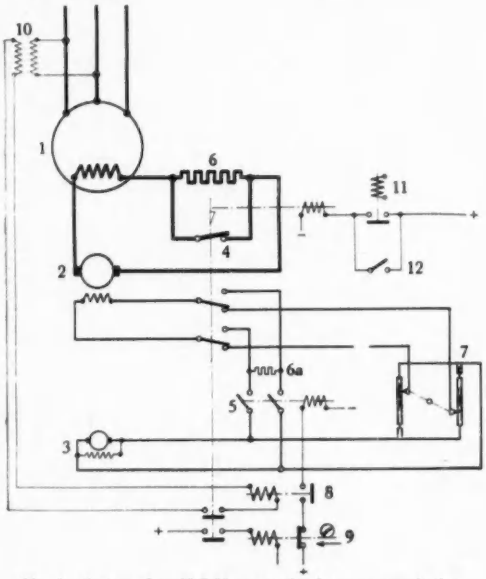

Fig. 4. **System of rapid field suppression by counter excitation.**
(1) Alternator.
(2) Main exciter.
(3) Pilot exciter.
(4) Field switch.
(5) Contactor for counter-excitation
(6, 6a) Field discharge resistors.
(7) Automatic voltage regulator.
(8) Instantaneous under-voltage relay.
(9) Time-lag relay.
(10) Voltage transformer.
(11) Protective relay for initiating de-excitation.
(12) Contact for initiating de-excitation by hand.

A serious short-circuit inside the alternator assists in speeding up field suppression. If the terminal voltage collapses completely, suppression is carried out without counter-excitation, since the under-voltage relay (8) cannot function. The inherent behaviour of the machine in this case assists rapid suppression, since the time taken to reduce the field is in any case less under short-circuit conditions than it is with de-excitation from no-load.

In order to compare various circuits for field suppression, Brown Boveri carried out exhaustive trials with a three-phase alternator rated at 44 MVA, 10·5 kV, 50 cps, 250 rpm. These trials included all the best-known methods of field suppression, the principal ones being the resistance method described earlier (Fig. 3), the Rüdenberg system of suppression by oscillation, which was previously recognized as the fastest method, and finally the method of rapid suppression employing counter-excitation. The trials were carried out under identical conditions for all systems and at various voltages. Similar results were obtained with each system, and are illustrated in Fig. 5, in which an extract from the oscillograms is reproduced. The initial shape of the curves is due to the over-voltage at the rotor during the initial stages, but, by choosing suitable discharge resistances, it can be kept roughly equal in all cases. What is noteworthy is that, despite the initial similarity in the voltage curves, i.e., with equally heavy voltage stresses on the exciter winding, the rest of the curve, down to complete disappearance of the alternator voltage, is quite different, depending on the method employed.

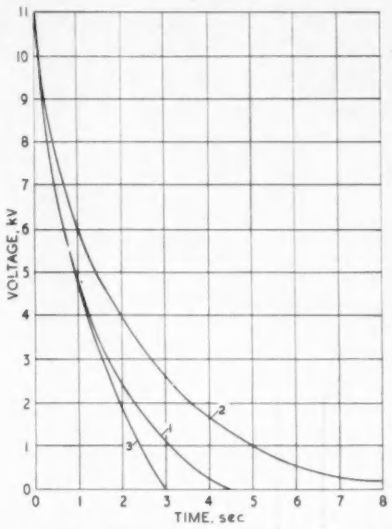

Fig. 5. **Curves showing reduction of the voltage of a three-phase alternator, using various methods of field suppression.**
(1) Rapid field suppression by the Rüdenberg system.
(2) Rapid field suppression with field switch and discharge resistance in rotor circuit, as shown in Fig. 3.
(3) Rapid field suppression by the Brown Boveri system of counter-excitation, as shown in Fig. 4.

Properties and Applications of Epoxy Foam Materials

(*Concluded from page 311*)

aircraft wings. In addition, because of their low water-absorption properties, they are suitable for use in such applications as rafts, life-buoys, and built-in flotation units for boats. It should, however, be appreciated that, at present, epoxy foam materials are more costly than competitive foam materials, so that, to justify the use of epoxy foams, which are still in the development stage for many applications, the particular properties inherent in these foams must be a definite requirement in a given application.

On the Wadkin heavy duty precision radial routers

These excellent machines depend to a great extent, for their superior performance, on the design of the barrel mounting and the roller carriage on the radial arm. The radial arm mounting, evolved by Wadkin Ltd. has straight Nitralloy tracks running in 16 special F.B.C. ball-bearing rollers mounted in the roller carriage. This in turn is bolted to the top of the rotating barrel, which is a heavily-ribbed casting. As shown in this section of the type H.Y.R., the barrel is carried at the top in a Timken tapered-roller bearing of light cross-section: this is 16¾-ins. bore. This bearing is heavily pre-loaded by a 4-ins. bore heavy-duty Timken bearing at the bottom end, thereby giving a 'rigid' mounting but with the necessary rotational freedom.

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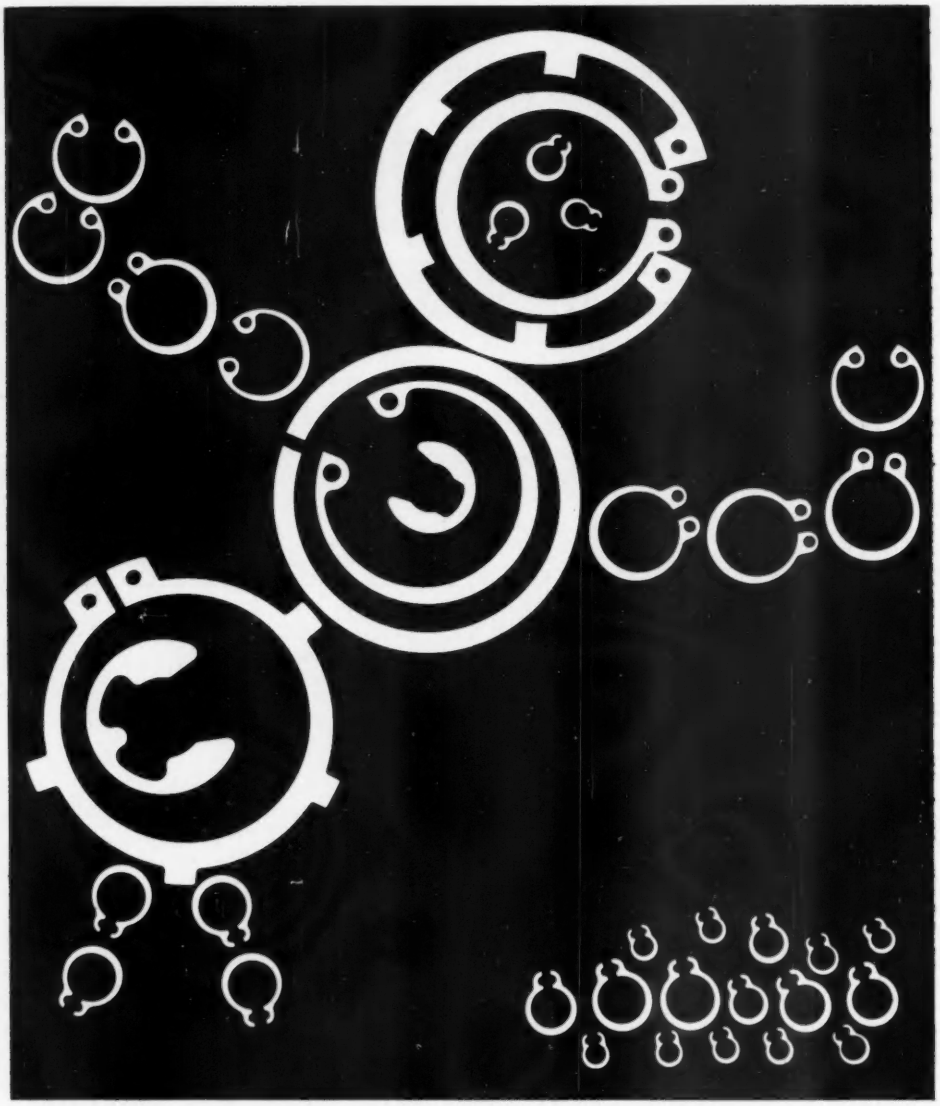

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British Industrial Developments

Combined Desulphurizing and Sulphuric Acid Plant

An entirely new combined desulphurizing and sulphuric acid plant, which is claimed to incorporate major advances on any existing similar process, has been developed by the Appleby-Frodingham Steel Company, a branch of The United Steel Companies Limited. The first full-scale plant is now being installed at the Appleby-Frodingham works and is scheduled for completion by mid-1959.

The new plant, which consists essentially of a fluidized hot iron oxide absorber and regenerator, is capable of removing both hydrogen sulphide and organic sulphur to very low levels ; at the same time, it permits the recovery of considerable quantities of sulphuric acid. The heat requirements of the plant are met by the combustion of the absorbed sulphur and, once started up, it is self-sustaining for heat. A pilot plant has been in successful operation for two years, purifying up to 3,000,000 cu ft of coke-oven gas per day. The full-scale plant, when completed, will deal with 32,000,000 cu ft of gas per day.

Because the process employs hot gas, it is very much faster than conventional iron oxide boxes or purification towers, while the use of fluidized beds reduces the amount of spent oxide by 80% and simplifies its handling and distribution. In compactness and capital cost, the process compares very favourably with existing methods, while the entire operation of the full-scale plant is capable of being handled by two men per shift.

Under the provisions of the Clean Air Act, the height of chimneys discharging sulphur dioxide to atmosphere may be subject to regulation and this, combined with the economic and technical factors, suggests that the new Frodingham desulphurization process is likely to be of particular interest to the steel industry, the gas industry (especially in the case of the larger gasworks), and the chemical and petroleum industries. Thus, for instance, it is estimated that the recovery of sulphur dioxide by the new process in an integrated steel works producing 1,000,000 tons of steel ingots per year enables as much as 250 tons of sulphuric acid to be produced per week.

Appleby-Frodingham have appointed two licensees to build plants based on this process, i.e., Henry Balfour and Company, Limited, of Leven, Fife, and W. J. Fraser and Company Limited, of Romford, Essex, who are building the unit at present under construction.

Controlled-Atmosphere Furnace for Heat-Treating Electronic Components

Designed and manufactured by Royce Electric Furnaces Ltd., of Walton-on-Thames, a new controlled-atmosphere package-type furnace for the bright heat treatment of photoelectric-tube components has been installed by the English Electric Valve Co. Ltd., of Chelmsford. The furnace provides closely controlled heat treatment at temperatures up to 1000°C in a hydrogen atmosphere and is operated by hand on a definite time cycle.

The charge is loaded on trays and is pushed progressively through preheat, heating, and water-cooled exit chambers, a continuous gas-tight duct containing the hydrogen atmosphere passing through all chambers from entrance to exit doors. Flame-curtains are provided at both doors and are arranged to supply gas automatically through a diffusion chamber as soon as the door is opened, preventing the entry of oxygen into the chamber and ensuring complete control of the furnace atmosphere with economy of gas consumption. Further economy of gas and the exclusion of deleterious action is maintained by the gas-tight duct.

Heating elements of heavy-gauge 80/20 nickel-chromium strip operate from the low-voltage secondary of a stepdown transformer, and are suspended in the heating chamber, external to the muffle, in such a way as to give free radiation to the muffle. Temperature control is by means of a transistor-type electronic controller operating in conjunction with the necessary switchgear, which is housed, complete with pilot lights and switches, in the furnace stand. A fusible-type thermal cut-out is fitted for automatically switching off the equipment in the event of accidental overheating.

An important feature is the control of the water-cooled exit chamber. Water-flow indicators give visual indication, and flow is controlled by immersion thermostats operating magnetic valves on the water supply. Indicators for measuring water temperature and gas-flow indicators for both the nitrogen purging gas and hydrogen operating gas are also fitted.

Can-Filling Machines for Frothing Liquids

Although capable of being used equally successfully with non-frothing liquids, the new Waddington-Duval can-filling machines announced by Liquid Systems Limited, of Croydon, are primarily designed for the rapid dispensing with extreme accuracy of highly frothing liquids. Unless otherwise specified, they are made of stainless steel, so that they can be used with most liquids and can dispense any number of different products.

In operation, the container to be filled is placed on a scale, which is set to the required quantity, with an accuracy of 0·5%. The supply is cut off instantly when the balance arm of the scale rises and interrupts a photo-electric cell. Avoidance of frothing is ensured by a patented filling lance, which is lowered to the bottom of the can before any liquid flows, and also by automatic control of the rate of flow in such a way that filling starts very quietly and continues evenly, thus avoiding turbulence. For non-foaming liquids, the operation can be adjusted for full-flow filling from the start, with a corresponding reduction in operating time.

Measuring 3 × 3 × 6 ft high, the machine fills containers with capacities of 1, 2, 3, 4, 5, and 10 gal. Changeover from one size to another takes approximately 10 sec, and a by-pass is incorporated to permit the filling of

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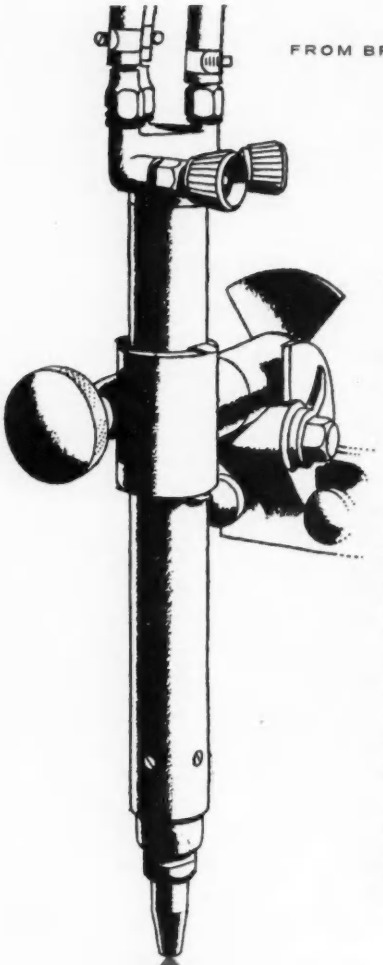

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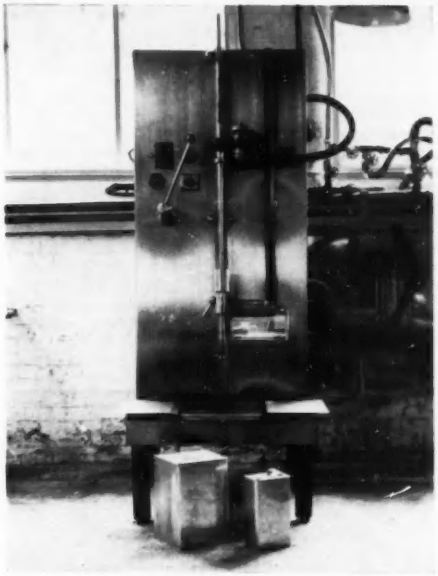

45-gal containers. The machine, which can be operated satisfactorily by unskilled labour, will dispense up to 60 tons per week of foaming liquid into sealed containers. Spillage is stated to be nil, and tests have shown that it is capable of saving from wastage 1 to 1·5% of all foaming liquid handled. Furthermore, it ensures that the outside of the container and the adjacent premises are kept dry and entirely free from contamination by the product, which is itself safeguarded from pollution.

New Machines for Speeding up Thermostat Production

The introduction of new types of machines developed by Smiths Motor Accessory Division has made possible a 20% increase in the output of thermostats from the Putney Vale factory of K.L.G. Sparking Plugs, Ltd., part of the S.M.A. group of companies. One such machine enables the metal bellows of thermostats to be spun and fashioned mechanically to a higher degree of accuracy than is possible by manual methods, while other machines are designed to deep-draw and "iron" brass stampings into thin-walled tubes, some 18 in. in length, from which the bellows are automatically formed. It is stated that this new method of drawing enables tubes to be drawn 10% thinner, i.e., at 0·0045 in., than was previously possible, resulting in a more sensitive bellows and consequently a more efficient thermostat.

The sequence of preparing the brass from which the bellows are manufactured is clearly shown in the illustration. Starting with a stamped-out disc, there follow three cup-forming and three tube-ironing processes which gradually increase the length of the tube, an outstanding feature being the small number of operations performed to produce the required length of tube to the necessary accuracy of wall thickness. Even more outstanding is the substitution of automatic spinning for hand spinning in the forming of the bellows, by means of which the machine takes the tube and, with a "finger" insert, pulls it outwards against a guide which forms a series of convolutions of predetermined length and diameter. Two bellows are spun in this way, in a matter of seconds, from one length of tube and, after each bellows has been formed to a previously calculated size, the machine cuts it off automatically.

To match the increased rate of output of bellows made possible by the new methods, semi-automation has also been carried to the assembly of the complete thermostats. A conveyor belt carries all the component parts to the operators of each sub-assembly point. The operator removes whichever item is required from the conveyor and solders it in position by using a specially adapted high-frequency soldering unit. This same conveyor ensures that all the components are perfectly clean. As each item comes away from the press or form tool, it is attached to the conveyor and, en route to the sub-assembly point, is passed through an electrolytic cleaning process.

From the moment the bellows come off their forming machine to the completion of the actual thermostat, very little handling of the components is evident, the assembly cycle being that the bellows are first spin-dried to remove surplus moisture and then annealed to relieve manufacturing stresses. The ends are cleaned and tinned on one of the many different rigs attached to the various high-frequency soldering units prior to having the bellows plates and valve stem soldered in position. As soon as this operation is completed, the unit so far assembled is ejected automatically from this rig and passed by a conveyor belt to the next sub-assembly stage, i.e., the fixing of the carrying bracket to the bellows. By means of an indexing table attached to the high-frequency soldering machine, the bracket is first soldered in position; a solution of methylated spirits and water (the thermostat controlling fluid) is then injected through a small hole in the bracket and into the bellows. Air is then evacuated from the bellows via this hole and, whilst still under vacuum, a taper pin is fed into the hole and pushed home prior to final sealing by soldering the pin in place.

The next stage is attaching the valve seating to the bracket, and this is achieved by a patented T-slot fixing. Whilst in position, the seating is trimmed to its correct size, and the valve is assembled. The completed unit is now placed on a special conveyorized calibrating rig, where it passes through water-filled calibration tanks, the temperatures of which are electronically controlled to ± 1°C. During its passage through these tanks the thermostat valve is finally soldered in position whilst still in water by another high-frequency soldering machine.

Testing is also performed on this conveyor. The thermostat is stressed to beyond its normal operating temperature by virtually boiling it. Its actual calibration is checked to within the required tolerance, and a check is also made to ensure that the valve is fully open 15°C above its calibrated opening temperature, as a safety precaution. From here the units are automatically ejected, visually inspected, and packed ready for dispatch at the rate of 25,000 per week.

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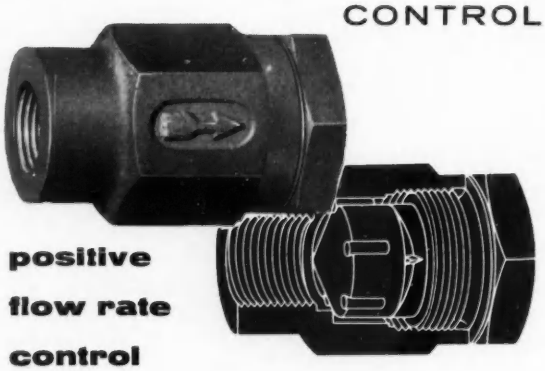

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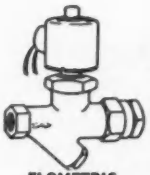

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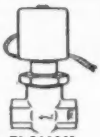

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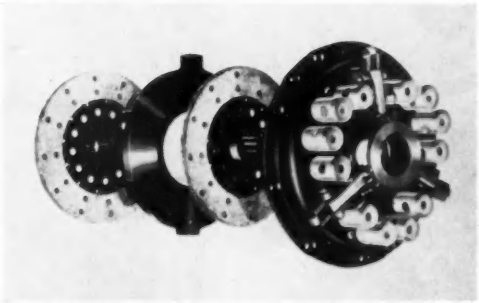
Integral Main-Drive Clutch and Power Take-off

The new Borg & Beck 10-in. integral main-drive clutch and power take-off announced by the Automotive Products Company Ltd., of Leamington Spa, is especially suitable for those applications where it is desired to provide independent control, from a single pedal, of the main drive to the wheels and of the power take-off. The unit basically consists of two separate and independent clutches mounted in tandem in a single robust pressed-steel cover, which is bolted to the flywheel in the usual manner. The cover contains the main and intermediate pressure plates, which are generously proportioned to aid heat dissipation.

A single ball-thrust release mechanism provides for the operation of both clutches. The clutch adjacent to the flywheel controls the drive to the power take-off, whereas the outer clutch operates the main drive through a hollow shaft, through which the power take-off shaft passes. The two clutch driven plates are similar, with the exception of the size of their splined hubs, which respectively engage the solid power take-off and the hollow



main-drive shafts, and are of the rigid type and fitted with heavy-duty woven metallic friction facings. With the pedal in the free position, both the main-drive and the power take-off clutches are engaged. When the pedal is depressed to an intermediate position, the main-drive clutch is disengaged, but full torque continues to be transmitted to the power take-off ; full depression of the pedal disengages both clutches, allowing the engine to run free of load. It will thus be seen that the unit provides for the simultaneous operation of the main drive and the power take-off, of the power take-off only, or of neither.

Maintenance of the clutch, which is of extremely compact design, is simple and straightforward, and is confined to two quickly performed operations. The first of these is the maintenance of a specified free travel at the clutch-pedal pad, and this is controlled by an adjuster incorporated in the pedal linkage in the normal manner. The second concerns maintenance of a specified clearance between three studs, mounted in the clutch cover and the outer or main pressure plate, and readily accessible through the clutch housing.

Indicator for Safe-Load Capacity of Fork-Lift Trucks

By dispensing with the normal rigid mounting between the moving end of the tilting jack and the mast structure, Lansing-Bagnall Ltd., of Basingstoke, Hampshire, have achieved a system of very accurate measurement of the moment of the load being lifted by fork-lift trucks, the results being transmitted to a large-diameter dial divided into "safe", "operate with care", and "danger" sectors.

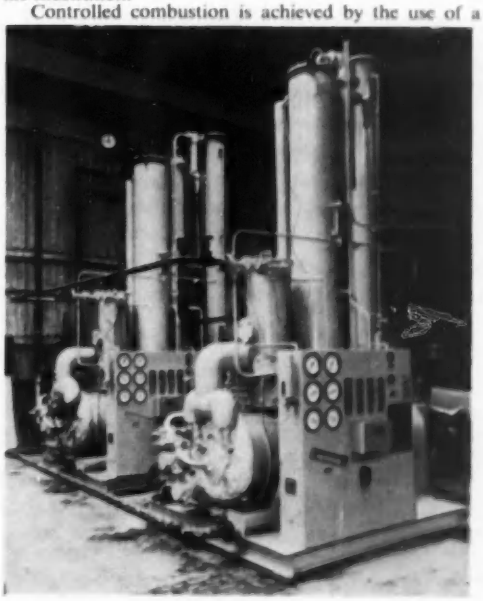
Essentially, the new system incorporates a point of leverage on each mast frame member, this pair of points

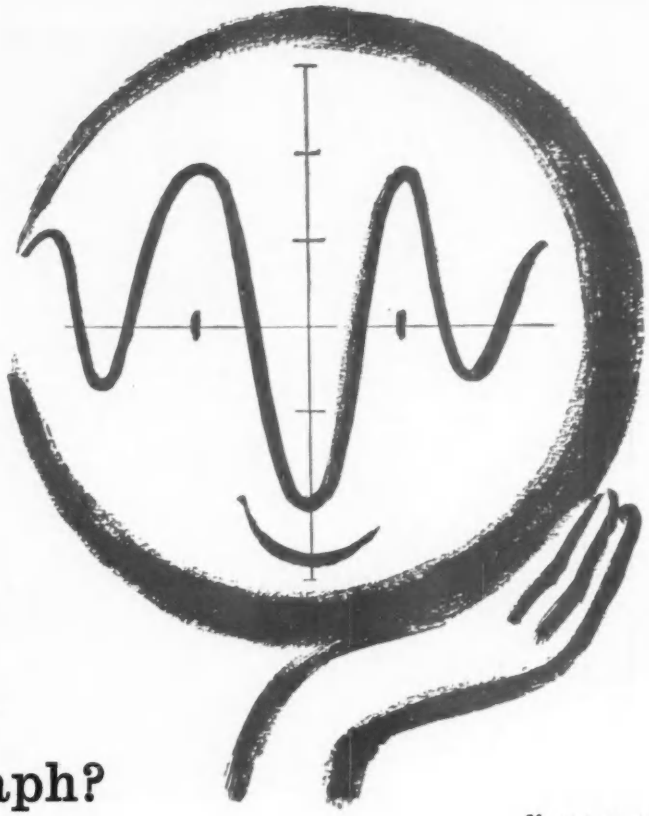
being connected to one central position, where an aneroid type of bellows measures the sum of the forces to which the tilting jacks themselves are being subjected. The subsequent movement of the bellows is then transferred hydraulically to the indicator dial. It is stated that, although very simple in principle, tests have shown the system to be remarkably accurate and capable of withstanding the rough treatment normally associated with industrial trucks. In addition, the mechanism is so sensitive that even a slight pressure of the fingers on the toe of the fork causes a noticeable variation in the position of the indicator pointer.

Nitrogen Generators

The Holmes-Kemp nitrogen generator announced by W. C. Holmes & Co. Ltd., of Turnbridge, Huddersfield, is capable of producing nitrogen by the combustion of any of the commercial fuel gases with air. Operation of this plant, which is available in capacities up to 20,000 cu ft per hour of nitrogen, can be semi-automatic, and the gas produced has a degree of purity which is stated to compare favourably with that of nitrogen made by any other full-scale production unit. The first two standard Holmes-Kemp generators to be manufactured in the U.K. at the company's Turnbridge works are shown in the illustration.

Controlled combustion is achieved by the use of a





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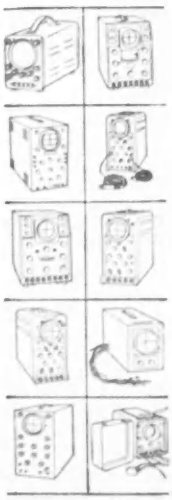

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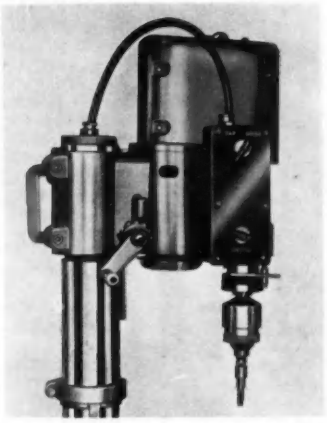
carburettor which is adjustable over a wide range to give varying fuel/air ratios and which is automatically responsive to demand. The products of combustion are cooled, and carbon dioxide removal is then carried out by passing the gas through an adsorber tower containing monoethanolamine. The plant is complete with reboiler, stripper, and heat exchanger for the regeneration of the monoethanolamine, thus providing a completely self-contained and continuous system of operation. A nitrogen compressor is usually provided to enable the gas to be stored at pressure. Where necessary, a Holmes-Kemp dryer can be included, to ensure freedom from moisture.

Automatic Reversing Tapping Attachment for Drilling Machines

As announced by the Mortimer Engineering Co., of London, N.W.10, the new "Ohma" automatic reversing device, originally introduced by AB Arboga Maskiner, of Arboga, Sweden, for their range of drilling machines, can now be fitted to practically any type of drilling machine.



This reversing device is a simple accessory which enables all thread-tapping work to be carried out by a drilling machine, without the need for any special tapping equipment. The change-over from drilling to tapping is effected by a simple switching arrangement, while tapping depth is set by means of a knob on a graduated scale. Depth setting is exact, and reversing is automatic.

Thermally Compensated Crystal Oscillators

A new type of thermally compensated crystal oscillator recently patented by the Automatic Telephone & Electric Co. Ltd., of London, W.C.2, is at present available for operation on any spot frequency between 4 and 16 Mcs and achieves a stability of ± 5 parts per million over a temperature range of -20 to $+70$ C with respect to any specified frequency. The high degree of stability is achieved (without resource to crystal ovens, thermostats, or other current-consuming devices) by a unique method whereby the oscillator, which is essentially of a conventional type, also has a temperature-compensating network controlled by a thermistor mounted inside the crystal envelope. The complete oscillator unit is available as a small sub-assembly which can easily be incorporated into a wide variety of equipment during the design stage. The current consumption (8 mA at 230 V d.c. and 0·3 A at 6·3 V) is no more than that of any other type of oscillator, and the units are ready for operation within a minute or two after switching on.

The advantages of this type of oscillator are expected to appeal particularly to designers of mobile VHF radio equipment, where increasingly higher stabilities over wide temperature ranges are required, owing to closer channel spacings. The compactness and stability of the unit are also expected to interest manufacturers of test equipment and, with this end in view, investigations are being made regarding the possibility of applying the same techniques to oscillators operating on lower frequencies, as well as those using battery-type valves and transistors.

The Effects of Irradiation on Magnetic Materials

(*Concluded from page 304*)

The results show that designers must not assume that magnetic materials will be generally satisfactory in devices that must operate in radiation environments with constant response. On the other hand, some materials are quite stable. Permendur appears to be promising for magnetic-amplifier cores subjected to radiation and heat, with the important drawback that its cobalt content leads to considerable induced activity.

Magnetic properties are insensitive to all types of radiations except fast neutrons. This suggests the possibility of making a magnetic fast-neutron dosimeter. In this connection, fast-neutron dose is indicated free of background from slow-neutron or gamma dose. Theory states that gamma photons produce only ionization effects, and slow neutrons produce impurity atoms only in a negligible number. When a 5–79 Molybdenum Permalloy core was irradiated with gammas alone ($1{\cdot}4 \times 10^{15}$ photons/cm^2, 1·3 MeV average energy), no change in magnetic properties was observed. The data imply the possibility of building a fast-neutron dosimeter, using the radiation effect on the magnetic properties of 5–79 Molybdenum Permalloy. This possibility should be investigated further.

Further Investigations

The results obtained suggest several additional experiments which would be useful and informative. In fact, the authors have already completed a second experiment in which ten other soft magnetic materials have been studied[5], while additional work has been reported by other investigators[6], and further research with permanent magnets is in progress.

On both 5–79 Molybdenum Permalloy and 16 Alfenol, diffraction and resistivity measurements should be made. This would permit correlation between changes in the degree of order produced by the radiation with changes in magnetic properties and resistivity.

Reheat treatment of an irradiated 4–79 Molybdenum Permalloy core has been carried out[5]. This re-annealing restored the pre-irradiation magnetic properties of this core, indicating that thermal-neutron effects (transmutations) are negligible, at least for this core.

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ADDITIONAL ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

Transverse Gauge Variation in Strip and Sheet Rolling.
By K. SAXL. (From an advance copy of a paper submitted to *The Institution of Mechanical Engineers, London*, May 1958, 13 pages, 20 illustrations.)

IN dealing with the problem of obtaining high economic production of flat strip from rolling mills, longitudinal gauge variation and its control has received considerable attention, but not so the transverse variation, though this is of no less importance. This variation, and particularly its redistribution during rolling, is considered to be the root cause of unsatisfactory flatness. Undesirable features such as locked-up longitudinal stresses, waviness of edges, buckling, edge thinning, edge cracking, and, in extreme cases, break-up of strip in the roll gap, are the visible signs of this variation or its unsuspected consequences.

Variation of the outgoing thickness is a measure of the non-uniformity of the roll gap during rolling which, in turn, is due to complex deformation of the roll surfaces, not adequately compensated by the cambers to which these surfaces are ground.

The mechanism of this deformation is accordingly investigated, its relation to the various defects encountered in rolling mill practice considered, and remedies suggested in this paper. A roll in a two-high mill deflects by bending and shear, and simple formulae for their determination are given. Its surface also deflects by local compression, and this is the cause of edge thinning in strip rolling. A method of estimating the magnitude of this thinning, based on a calculated value of the compression of the rolling surfaces, is suggested.

Conditions in four-high and other backed-up mills are more complicated. The comparatively slender rolls are supported by backing rolls, but the support is not rigid, owing to the compression of the two curved surfaces. The effect of this elastic support in certain circumstances is to bend the work roll to an extent which makes the lateral flexibility of the backing roll so small in comparison as to be almost irrelevant. A four-high mill with slender work rolls is liable to suffer from this difficulty, particularly when rolling a comparatively narrow strip. A method of cambering designed to overcome it by shortening the effective support between the work roll and the backing roll is accordingly suggested. This can be arranged to correct the edge thinning, as in two-high mills. Its chief value does, however, rest in preventing, or at least reducing, the secondary deflection of the work rolls.

The secondary deflection of the work rolls can be partially compensated by exerting a controllable force on their chocks, bending the necks of the work rolls towards their respective backing rolls. Formulae for calculating the desirable force are suggested. This form of control will be more effective if employed in conjunction with the shortened support of the work roll on the backing roll.

It is shown that change of thickness distribution across the strip during a pass must result in considerable lateral forces, which tend to bend the work roll, buckle the strip, and are, in fact, responsible for a number of further petty troubles in strip rolling. Lateral bending of strip after slitting is another demonstration of the considerable internal stresses due to this factor. It is suggested that offsetting of the work roll towards the outgoing side, which is favoured by some operators, is a palliative. The cure is gauge uniformity. It is shown that variation in thickness distribution will create conditions conducive to edge cracking and may, indeed, be its primary cause in certain circumstances. One of the implications arising out of this investigation is the necessity to maintain predetermined conditions, particularly the width and roll-separating force.

The rolling load will be much less critical in four-high mills with rolls cambered as suggested, and this enables higher loads and better utilization to be achieved without impairing the product.

Cutting Tools made from High-Speed Steel Chips.
By YU. P. ZIMIN and K. M. STROYEVA. (From *Stanki i Instrument, Russia*, No. 2, 1958, page 15, 2 illustrations.)

As a result of the successful compacting of alloy-steel swarf into satisfactory forgings for use in making machine components, the method has been developed to include compacting chips of high-speed steel for the manufacture of cutting tools.

As described in this article, a steel cylindrical vessel, about 1⅜ in. in diameter and 5¼ in. in height, was tightly packed by hand with chips obtained by milling two high-speed steels, i.e., R18 (containing 0·78% C, 18·6% W, 4·15% Cr, 1·2% V, 0·34% Mn, and 0·32% Si) and R9 (containing 0·9% C, 9·5% W, 4·0% Cr, and 2·3% V). The charge in each case was compressed with a forging hammer and was then heated in a furnace to from 1100 to 1150°C for from 15 to 20 min, and then placed in a preheated die. The first two or three hammer blows (900 lb) compressed the chips to a height of 1¼ to 1½ in., and subsequent blows (2900 lb) produced solid forgings of ¾ to 1 in. in height, the close contact between the chips, their large area, and the heat generated in plastic deformation ensuring solidity and strength, without any apparent disruptions.

Chemical analysis of the forgings obtained showed only slight differences in the composition of the steels concerned. Thus, in the case of the R18 steel, the composition after forging was 0·71% C, 18·55% W, 3·8% Cr, 1·0% V, 0·32% Mn, and 0·14% Si. Basically similar results were obtained with the R9 steel. Chip boundaries could be observed in polished micro-sections, and some carbide non-uniformity and segregations were noticed, but these were within prescribed limits. The Brinell hardness of the forgings after isothermal annealing was between 230 and 255, as compared with 295 to 355 before annealing, differing little from that of the parent steel.

Tool tips were then made with samples of R9 and R 18 steels, both in their parent form and in the forged-chip form. The results of these tests show that cutting tools made from high-speed tool chips possess practically the same cutting properties as tools made from the parent metals.

Ultraphotometer using a Magnetically Modulated Photomultiplier.
By G. O. STRIKER. (From *Acta Technica Academiae Scientiarum Hungaricae, Hungary*, Vol. 20, Nos. 1–2, 1958, pp. 83–102, 15 illustrations.)

THE measurement of small light intensities is constantly gaining importance, and more and more physical and chemical properties of materials are being determined directly or indirectly with light as the measuring medium. There is a growing need for more sensitive, yet reliable and stable, methods of measuring low light fluxes, and the importance of such methods is increasing with the widespread use of quantitative X-ray and nuclear radiation measurements, using phosphors and scintillating crystals as energy transformers.

The object of the investigation described in this paper was to develop a new type of highly sensitive photometer meeting industrial requirements, using the principle of

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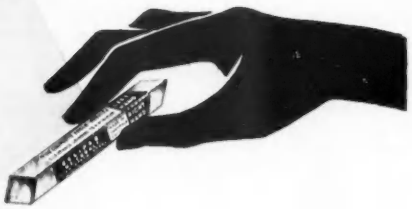

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magnetic modulation to photomultipliers. Theoretical considerations indicate that, by applying this principle, known as the Magnephot principle, to photomultipliers, a simple and rugged industrial microphotometer can be built in the range of extremely low light intensities. The requirements for a low input impedance and a conservative input signal level can be successfully combined with a satisfactory signal-to-noise ratio, without the use of mechanical light chopping. The external-shield method of dark current suppression can be usefully employed to extend the sensitivity limit to extremely low light values, limited only by the discontinuous nature of the light flux.

Experimental investigations have confirmed these assumptions and have led to the construction of a sturdy and simple ultraphotometer of inherent electrical stability. The limit of useful sensitivity is around 10^{-12} lumen, corresponding to the light collected by a 24-in. telescope from a star of the 14th order. Experiments with scintillating crystals yielded data about the use of the new instrument as a quantitative sensing device for nuclear radiation.

The experimental instrument gave satisfactory performance as to zero stability, independence from leakage currents, and the exclusion of dark current without cooling. The use of a.c. amplification assured convenient range-switching and the possibility of amplifying the signal further for registration or control purposes. The new ultraphotometer will now be investigated in conjunction with spectrophotometry, and astronomic and other low-level photometry, as well as the indirect measurement of X-ray and nuclear radiation.

The Mechanical Design of an Experimental Plant Type of Gas Turbine

By L. T. WHITEHEAD. (From an advance copy of a paper submitted to *The Institution of Mechanical Engineers, London*, May 1958, 22 pages, 24 illustrations.)

PROBLEMS of thermal expansion and insulation encountered in the structural design of steam turbines are accentuated in gas turbines, owing to their generally higher operating temperature, and various technical papers have been published over the past ten years in connection with the mechanical construction of gas turbines for applications other than aircraft, from which a distinct cleavage of opinion emerges.

On the one hand are the protagonists of weight, heavy sections, and general robustness as essentials for preserving alignment and running clearances. This view owes much to steam-turbine tradition, and existing examples of this practice are, in fact, generally the products of firms well established in that field. On the other hand, there are those concerned mainly with aircraft gas turbines who have achieved very high power/weight ratios on units which can run up to full load in a short time. Both these advantages are attributable to the avoidance of the heavy section which the opposite camp prefers.

Weighing the evidence, it appears reasonable to look more towards aircraft turbine practice when considering the higher temperatures for industrial or similar applications. However, one important point to be remembered is the very different length of life required, e.g., about 500 hr for the aircraft engine and up to 100,000 hr for the power-station turbine. These are the considerations which have influenced the mechanical design of the turbine discussed in this paper, which describes the construction of a four-stage, 8000-bhp gas turbine designed and manufactured at the National Gas Turbine Establishment, and embodying several features offering a solution of the problems involved.

A tubular rotor reinforced by radial discs is required if material is to be used with economy, and, in addition, the rotor must be maintained at a temperature which allows full use of the material properties. Heat should be confined as closely as possible to the gas-flow annulus to obtain the maximum output of work from the blading and to avoid undesirable thermal stress in the rotor. The insulation of the bearings and the provision of some flexibility between these and the shaft, to meet the possibility of damage from excessive heat soakage in an emergency, are also important. The methods used to attain the above objectives include rotor discs connected by threaded side-rims of large diameter and cooled by an axial flow of air under the blade roots.

Flexible sleeves are fitted between the shaft journals and the roller-type bearings, to inhibit heat flow and to accommodate differential radial expansion. The stator casings are treated as gas-carrying ducts, relieved of mounting constraints and bearing-alignment duty by being freely mounted on radially disposed pegs and keys, giving freedom for thermal expansion in all directions. Preliminary experiments and rig tests on the bearing mounting sleeves, and on surface treatment, to prevent seizure of the screw-thread connections between the rotor discs, are described. More than usual attention is given in the design to methods of facilitating inspection and maintenance, and provision is made for replacing individual rotor blades and bearings without separating rotor and stator.

Cooling Rates and Peak Temperatures in Fusion Welding.

By C. M. ADAMS, JR. (From *The Welding Journal, U.S.A.*, Vol. 37, No. 5, May 1958, pp. 210s–215s, 10 illustrations.)

IN the field of welding metallurgy, much attention is given to the effects which a welding arc or flame will have on the structure and properties of the metals being joined. To gain some quantitative appreciation of the way in which the metallurgical effects of welding depend on welding variables and on geometric and physical properties, considerable attention has been given to the unique heat-transfer phenomena associated with the movement of heat sources in or on metal surfaces. The mathematics of conduction from moving heat sources has been treated in classical works on heat conduction, and, with the establishment of certain idealized boundary conditions, solutions presented can be applied quite realistically to the field of welding.

Most of the solutions to be found in the literature on heat conduction have been presented in terms of the distribution of temperature around a moving point or line source of heat. However, the instantaneous temperature distribution around a point source does not lead immediately to an understanding of metallurgical consequences which may result from passage of the heat source.

To obtain metallurgically useful information, numerical solutions of the temperature equation are required, in order that a continuous curve of temperature against time at a particular point near the weld may be derived. From studying such a curve, the peak temperature experienced at the point in question may be determined, and theoretical heating or cooling rates have usually been obtained by measuring the slopes of such curves. To arrive at these values, directly and numerically, starting with the temperature-distribution equation, is an arduous and indirect task, and it is the object of this paper to present engineering relationships affording the direct calculation of peak temperatures or cooling rates from welding variables and thermal properties.

It has been found possible to derive relationships giving peak temperatures and cooling (or heating) rates directly as functions of geometric, thermal, and welding variables. The expressions are close approximations, derived from the published temperature equations, demonstrably accurate to within 5%, and are simple enough to be of engineering usefulness. The most significant feature of these simplified equations is the clarity with which effects of changes in any welding, thermal, or geometric variables can be predicted. Many results depend on whether the object being welded behaves like a thin sheet (two-dimensional heat flow) or a thick plate (three-dimensional heat flow), and this is readily determined, as shown in this paper.

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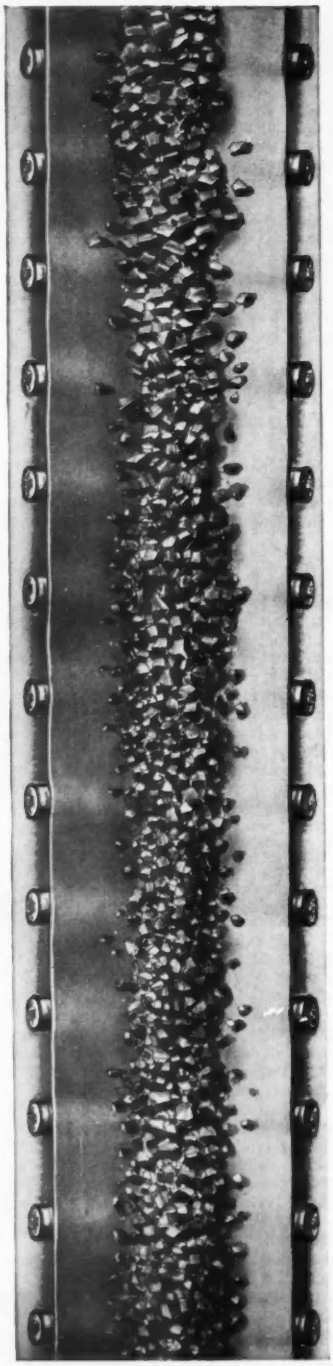
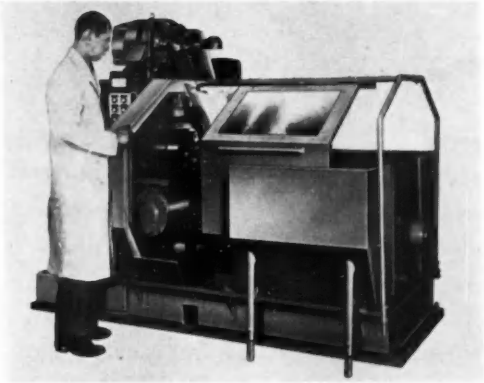


New Materials, Processes, and Equipment

HIGH-SPEED AUTOMATIC TURRET LATHE

A new automatic heavy-duty turret lathe, designated model 3E-15, is announced by the Potter & Johnston Company, of Pawtucket, Rhode Island, a subsidiary of the Pratt and Whitney Company. Designed specifically for high-speed production of hard-to-machine alloy components, the new machine has a six-face turret, a 10-in. chuck capacity, and an 11½-in. swing over the base-ways, and will hold close tolerances, produce smooth finishes, and handle a wide variety of work in the small- to medium-size range.

The lathe is equipped with an entirely new 15-hp headstock, all-electric speed clutches, and precision-ground speed gears. To withstand the strains of hogging-out tough metal rapidly with heavy feeds under full power, construction is unusually robust. Independent front and rear cross-slides are individually adjustable along the base-ways. In addition, a new control mechanism permits setting of the front or rear slide for on-time or delayed movement relative to any turret face merely by turning a selector switch.



Since there are no cams to change, set-up time has been substantially reduced. As a result, production runs of as few as 25 pieces can be economically handled on a fully automatic basis. Other features contributing to this shortened set-up time are an oversized, easy-to-set programming drum and readily-accessible change gears. Four automatic speed changes and six automatic feed changes for each set of change gears are provided.

CHEMICAL METHOD OF JOINING NON-FERROUS METALS

Stated to produce joints which represent a totally new concept in the union of metals, as they cannot be classified as soldering, welding, or brazing operations, a new material, designated " InterAct ", for the chemical union of similar and dissimilar non-ferrous metals has recently been developed by Intertectics, Inc., of Cleveland, Ohio. The new material makes it commercially feasible, for the first time, to join copper and aluminium by chemical means, as distinct from the mechanical means hitherto employed. In addition, it forms a permanent chemical bond through ion exchange between such materials as magnesium, titanium, brass, zinc, silver, and gold. The resulting joint is stated to be stronger in most cases than either of the metals involved and to have higher corrosion resistance and similar electrical, physical, and chemical properties.

Basically, " InterAct " is a material which, under the influence of heat, eliminates the surface oxide layer and causes similar and dissimilar metals to flow together at temperatures lower than the melting points of either metal used. As such, it may be called a reaction eutectic. It is at present available in three forms, i.e., InterAct-S, InterAct-E, and InterAct-G. InterAct-S is composed of chlorides of zinc, lithium, potassium, and sodium, encased in an extremely pure alloy of zinc and aluminium, and supplied in coils, on which ⅛-in. o.d. wire is wound. Both the material and the sheathing play an integral part in the joining process. In this connection, it should be understood that the material itself is highly hygroscopic and exhibits corrosive characteristics which, on contact with moisture, would quickly destroy itself and its container. This difficulty is overcome by means of the special metal container or sheath, which is formed by a special method as the InterAct is injected under vacuum, so that the container is completed the moment the material has been placed in the shell. The ends of the coil are sealed with polyethylene caps, so that the shelf life of InterAct-S is indefinite. InterAct-S is recommended for T-joints, butt joints, and other exposed-type junctions.

InterAct-E has the same formula as InterAct-S, but is supplied in powder form, sealed in polyethylene bags, and is recommended for joining wider contact areas. InterAct-G is composed of aluminium trichloride, zinc chloride, and sodium chloride, and is supplied in granular form, sealed in polyethylene bags. InterAct-G is recommended for joining foils and other materials where large surfaces are to be brought together at temperatures lower than those used with InterAct-E. InterAct-E and InterAct-G are also highly hygroscopic, so that their shelf life is indefinite only if the polyethylene bags are unopened.

One of the chief features of the new material is that it lends itself well to automated industrial conveyor techniques. With InterAct-S, three basic methods of use are possible. Of these, one method involves manual feeding of the InterAct wire into the joint, while the latter is being heated by a torch. Once reaction temperature has been reached, the reaction is completed in approximately two seconds, after which the joint is washed with warm water to remove reaction by-products. Under these conditions, the operator must be masked, as the material is toxic to the skin and lungs.

Another method is intended for conveyor operation, in which the materials to be joined are moved on a conveyor through an induction furnace or gas oven, heated to a temperature of about 810 F. The InterAct material, formed in ¼-in. diameters to fit standard industrial welding guns, is then fed through a mechanized gun to the joint, reaction being instantaneous. A water spray is then used to clean the joint. A hooded exhaust system, located over the junction point, vents fumes to the outside and, as these fumes are soluble in water, a water spray in the tower renders them harmless before their final escape to the exterior.

A third industrial method is similar to that just mentioned, except that, by means of electrical strip heaters, only those areas to be joined are heated. This method lends itself particularly to preforming of the wire coils to fit unusually shaped joints. The preformed coils can be dropped automatically into place on the joint in a conveyor-type operation.

It would appear that the material should be especially suitable for use in the construction of aircraft and guided missiles, where the joining of aluminium, copper, and titanium is a problem, and also in the electrical manufacturing and appliance industries, where it should eliminate high-resistance welds and complicated casting techniques.

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UNIVERSAL GRINDING MACHINE WITH SPECIAL TOOL-GRINDING ATTACHMENT

A new universal grinding machine with a special tool-grinding attachment, announced by the George Gorton Machine Company, of Racine, Wisconsin, permits accurate grinding of a wide variety of end-milling, die-sinking, and engraving cutters. Single- or multiple-flute, square, conical or ball nose, and straight or taper cutters with either straight or spiral flutes, of any angle, radius, or diameter, and any degree of clearance can be finished.

The attachment, designated " Toolgrind ", is completely universal, and incorporates a tilting spindle, compound cross-slides with micrometer dials graduated in thousandths of an inch, and an adjustable tooth rest, in conjunction with a micrometer dial for multi-flute grinding. The base of the device is graduated 360 deg. and has an index dial with a plunger providing for from one to six flutes. A dust-sealed anti-friction bearing spindle and collet sleeve are provided, together with adjustable stop dogs, a pivot-centre locating arm, and positive adjustments for wear. The " Toolgrind " attachment can be seen in the illustration at the top rear of the machine, together with a rack at the front holding an assortment of the end mills provided.

ELECTRONIC MICROMETER

An electronic micrometer, representing a unique method of measurement of small distances in the range of 250 to 40,000 micro-inches with very high accuracy, has recently been developed by The Wayne Kerr Laboratories Limited, of Chessington, Surrey. In principle, the new micrometer, designated type B.721, measures displacement or distance in terms of the capacitance change between the surface of a testpiece and non-contacting probes brought up to it. In other words, a capacitor is formed between the surface of the testpiece and the probe, the spacing of the plates of the capacitor determining the capacitance value.

The capacitor formed in this way is connected to a transformer-coupled bridge and is compared against an internal standard capacitor built into the bridge network and formed by a parallel arrangement of a fixed plate surrounded by suitable guard electrodes and a movable plate (earthed) driven by a precision micrometer. The transformer is accurately wound to provide two ratios, i.e., 1:1 and 10:1, giving a wide range of distances which can be measured. The standard capacitor and probe are made with very high precision to produce accurately a ratio of 100:1 in capacitance. In addition to the bridge network, the instrument contains a stable oscillator operating at a frequency of 10,000 cps and supplying the bridge with the signal, a high-gain amplifier tuned to the same frequency, and a phase-sensitive detector followed by a null-indicating meter. An alternative model, used mainly for precision research investigations, uses a multiple magic-eye indicator.

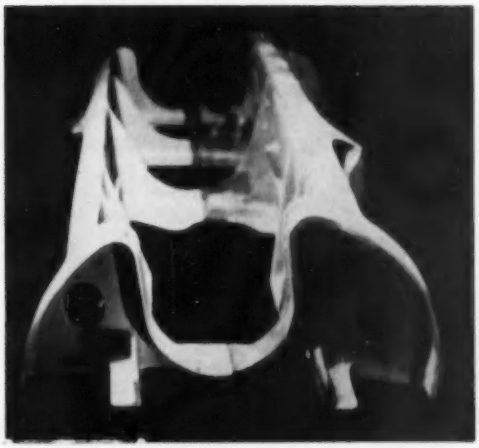
Operation is extremely simple, the only control used being the micrometer drive for setting the standard capacitor to obtain zero balance. The probe (connected by a cable to the bridge) is brought up to the metallic surface of the testpiece, without actually contacting it. The bridge is then balanced for null with the micrometer drive, and the distance is then read directly from the micrometer scale. The testpiece can then be raised to a given temperature to determine temperature coefficient, subjected to stresses to determine its modulus of elasticity or rigidity, rotated at high speeds to determine dilation of rotating parts, or replaced by another sample which is to be compared with the original, i.e., in the case of grading of rollers, ball bearings, etc. Other applications of a similar nature are, of course, also possible.

The instrument is of convenient size and weight, making it particularly suitable for use on location. Facilities are provided for feeding the information received into a recorder, in cases where continuous monitoring is required, and a variety of probes for different applications is available. The fact that no direct contact with the testpiece is required is a particular advantage, and is of especial importance in the case of fragile samples and rotating machinery. Furthermore, as measurements can be carried out at distances up to 70 ft from the testpiece, absolute safety is provided for the operator when performing tests on rotating machinery, as the instrument can be located outside the test chamber.

CHEMICAL MILLING OF MAGNESIUM-THORIUM ALLOYS

Recent developments in chemical-milling techniques in connection with magnesium-thorium alloys in sheet, extruded, and cast forms, as announced by the United States Chemical-Milling Corporation, of Manhattan Beach, California, are stated to simplify machining processes in the fabrication of aircraft and missile components made from these middle-range, high-temperature-resistant alloys, and to eliminate health hazards arising from their mild radioactivity.

In conventional methods of machining magnesium-thorium alloys, health hazards from the radioactivity of their thorium content are mainly limited to internal radiation from inhalation of dusts produced during grinding operations, or of fumes generated during welding and pickling operations. In the case of chemical milling, no dusts are produced, and fuming is negligible, while the disposal of any incidental wastes accumulated



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* 30 lbs. of dry powder, pressurized with 11 ozs. of carbon dioxide which expels the contents at the rate of 1 lb. per second for 28 seconds. * The powder is non-toxic, non-conductive of electricity, non-abrasive, frost proof down to −40°F., and harmless to machinery. * Easy to operate for untrained personnel, used in the sensible upright position and almost instantaneous in action. * The powder cloud has great heat-shielding properties, enabling even an amateur to tackle a serious fire effectively. * Easily installed, the Model 1630 also reduces maintenance costs. A tell-tale gauge provides instant visual confirmation that the extinguisher is under pressure.

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during the process is accomplished by standard methods which are generally acceptable in the handling of chemical solutions. Furthermore, as is the case with other magnesium and magnesium-alloy materials, the chemical milling of magnesium-thorium alloys is free from the combustion hazards which normally complicate the physical machining and welding of magnesium-base materials.

The process can successfully be applied to simple or integrally reinforced skins and structural members, either straight or tapered, and in large sizes or numbers, without essentially increasing the complexity of the basic tooling. The illustration shows an example of the application of chemical milling to a magnesium-thorium casting, in this case a structural support for the exterior surface of a control element.

HIGH-TEMPERATURE GALVANOMETERS

Basically intended for use with their Type 5-122 recording oscillograph for high-altitude, high-temperature flight testing, six new high-temperature galvanometers are announced by Consolidated Electrodynamics Corporation, of Pasadena, California. Stated to be the only ones of their type at present available for operation at temperatures as high as at least 175°F, these new instruments incorporate special lens mounts, mirror coatings, and extra precautions in soldering joints, enabling them to operate satisfactorily at high temperatures without any danger of corrosion or damage.

In two of these galvanometers, fluid damping is used, the other four being damped electromagnetically and capable of operating, with minor changes in damping and sensitivity, up to 250°F for extended periods of time, as against 200°F for the fluid-damped instruments. A typical example of one of the new range of galvanometers is shown in the illustration.

PORTABLE TRANSISTORIZED AMPLIFIER

Measuring only 8 × 3½ × 6 in. high and weighing 5½ lb, a new portable transistorized amplifier announced by Pye Telecommunications Limited, of Cambridge, is particularly suitable for use wherever a loudspeaker system is necessary, e.g., for the amplification of speech or music, but where the installation of a conventional and more complicated amplification system is considered impracticable, especially when a mains supply of current is not available.

Operating on a 12-V power supply derived from dry batteries or a small accumulator, the new amplifier, even at its full output of 10 W, has a current consumption of only 1·8 A, i.e., less than 20% that of a standard non-transistorized amplifier of comparable output rating, with, of course, the added advantage of greater compactness and portability. A wide range of suitable microphones and loudspeakers for use with this amplifier is also available.

HIGH-SENSITIVITY A.C. VALVE VOLTMETER

Extremely compact and robust, the new type-77 a.c. valve voltmeter announced by Advance Components Limited, of Ilford, Essex, is a highly sensitive instrument, capable of making accurate measurements from 100 microvolts to 300 V in twelve ranges, and utilizes a circuit which provides readings close to r.m.s. values, even when the harmonic content is comparatively high.

The instrument, which incorporates a four-stage amplifier with bridge rectification to the meter, has a scale calibrated in r.m.s. volts and also decibels, from —12 to + 2dB (relative to 1 mW/600 ohms). Measurements may be made at frequencies between 10 cps and 5 Mcs and, as the instrument has a high input impedance (10 megohms), it is useful for all amplifier and voltage measurements generally. It can also be used as a null detector, indicator, or amplifier for frequencies between 10 cps and 10 Mcs, and as an amplifier with a gain of 1000, adjustable in 10-dB steps, up to 5 Mcs. High impedance loads do not affect the voltmeter reading. A low-capacity lead, which completely screens the instrument, is provided, and incorporates a screened test-probe and a device which enables measurements to be made with an input capacity of less than 5pF.

The instrument, which is mains-operated, has a 4½-in. scale, fitted with an anti-parallax mirror. A slow-blow fuse device permitting temporary short-circuits is fitted. The application of an incorrect high voltage at the input terminals cannot damage the meter movement or the rest of the instrument.

HIGH-TEMPERATURE PHENOLIC RESIN FOR AIRCRAFT AND MISSILE COMPONENTS

A new phenolic laminating resin has been developed by the Barrett Division, Allied Chemical Corporation, of New York, for the fabrication of lightweight radomes, missile heads, and structural aircraft components of complex shape. It is stated that parts fabricated from this material, designated "Plaskon" V-204 laminating varnish, will retain their high strength at temperatures between 500 and 600°F and that, in fact, its strength at elevated temperatures exceeds that of the chief competitive phenolic materials formerly utilized in such applications.

Glass-fabric laminates impregnated with the V-204 resin can be made in standard multi-opening presses, using conventional plastic laminating techniques. In addition, glass-fabric materials impregnated with the new resin are readily adaptable to the fabrication of complex-shaped parts by a modification of the conventional vacuum-bag moulding process, this modification being necessary in order to permit removal of volatile substances and the water of condensation generated by the cure of the phenolic resin. Vacuum-bag moulding at from 12 to 14 psi, laminating at pressures as high as 2000 psi, or matched-metal die-moulding at even higher pressures are all feasible with materials impregnated with the new laminating varnish.

what, no nuts? . . .

Nuts to use nuts? Well, not exactly, but the fact that the Tapping Screw obviates the use of nuts is just one of its distinct advantages over the conventional Machine Screw. There's the saving in time to consider too. And cost. And there's no doubt whatever that the tapping screw makes for a better job all round. With its self-locating point the tapping screw forms its own thread thereby eliminating a separate and expensive tapping operation. Obviously it makes for easier assembly, is stronger, cheaper and faster. Safer too when the Phillips recessed head is specified, designed for power driving without damage to worker or job. You'll be well advised to change over to Tapping Screws. Nothing nuts about that !

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NEWS OF THE MONTH

BUSINESS NOTES

Albright & Wilson (Mfg) Ltd. announce that they are marketing a new process for bright nickel plating. This process was originally developed in the U.S.A. by the **Hanson-Van Winkle-Munning Co.**, where it was known as " Nickel-Lume " and " Levelume ". Albright & Wilson have applied for registration of the name " Plusbrite " for this process, which is completely new to the U.K.

To meet the increasing demand for ferrites in the computer industry, the Chemical and Metallurgical Division of **The Plessey Company Limited,** of Towcester, Northants., has recently moved a section of its plant engaged on the production of magnetic memory planes and stores to a new factory located near the parent factory at Water Lane, Towcester.

Roto-Finish Limited, of Hemel Hempstead, Herts., have acquired from Jacquet-Hispano Suiza, S.A., the controlling interest in **Modern Electrolytic Patents and Processes Limited,** who specialize in the electropolishing of metals. The closest technical liaison will be maintained between Roto-Finish Limited and M.E.P.P. Ltd., and a joint service will be available to determine the most suitable and economical process for deburring or polishing components.

F. B. H. Durachrome Ltd., of Kings Norton, Birmingham 30, announce that extensions to their existing hard-chrome-plating plant have now been completed, resulting in an increased production capacity of 100%.

Specialloid Limited, who have acquired the major share capital of **Powder Couplings Limited,** are now commencing the manufacture of powder couplings of a design which differs substantially from those manufactured by **Stone-Wallwork Limited.** Ball and roller bearings will be used instead of bronze bushings, and mechanical seals will replace the felt type. Full details can be obtained on application to Powder Couplings Limited, 20 Black Bull Street, Leeds 10.

Sheepbridge Alloy Castings Ltd., a member of The Sheepbridge Engineering Group, announce that **E. H. Bennet & Son,** of 124 Seymour Place, London, W.1, have been appointed their Southern area representatives.

Honeywell-Brown Limited have changed their name to **Honeywell Controls Limited,** and all head-office departments and the London branch office have moved from Perivale to a new building in Ruislip Road East, Greenford, Middlesex (Tel.: Waxlow 2333).

A new fuel-flow test-house, the most extensive of its kind in Europe, has been set up at Treforest, Cardiff, South Wales, by **Firth Cleveland Instruments Ltd.,** the recently formed subsidiary of **Simmonds Aerocessories Ltd.** The test-house provides facilities for calibrating and production-testing flowmeters, water separators, valves, and other fluid-handling equipment. It includes modern apparatus for future development requirements, and complete facilities are available for testing to Government and other official specifications to all manufacturers of equipment concerned with liquid-fuel products, both in the U.K. and in Western Europe.

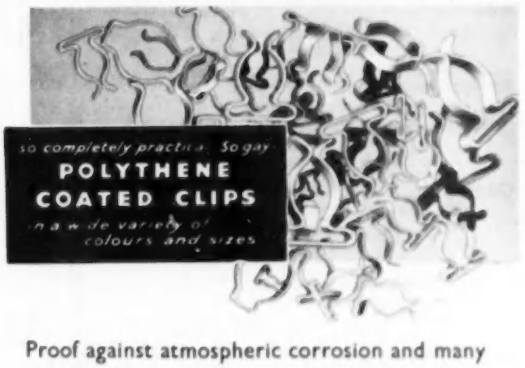
James Hodgkinson (Salford) Limited have purchased the ordinary share capital of **J. & J. Neil (Temple) Limited,** combustion engineers. The board of this company now comprises Mr. J. Douglas Cooper (chairman), Mr. N. C. Crighton, Mr. E. L. Hollinshead, and Mr. W. Murray (managing director). Continuity of the business of J. & J. Neil (Temple) Ltd. is thus assured.

A new British company, **Landis Machine-Maiden Limited,** has been formed by the combination of the **Landis Machine Company,** of Waynesboro, Penn., U.S.A., and **Naiden & Co. Limited,** of Hyde, Cheshire. The factory space at Hyde will be extended by a further 15,000 sq ft, thereby nearly doubling existing factory space, and, in the initial stage, plant and equipment will be installed for the production of tangential chasers. The products of both companies are complementary, and the new company will continue to build threading machines in the current " Maiden " range, as well as Landis tangential chasers and other Landis products in the field of thread cutting and thread rolling.

A new company, **Dowty Mining Developments Ltd.,** has been added to the Dowty Group of companies, and will carry out forward planning for mining development on behalf of **Dowty Mining Equipment Ltd.,** who have for years supplied hydraulic pit-props and self-advancing supports to mines all over the country. The new company will be based at Ashchurch, near Tewkesbury, also the home of Dowty Mining Equipment Ltd.

Aero Research Limited has changed its name to **CIBA (A.R.L.) Limited.** However, the trade-names of " Aerolite ", " Araldite ", " Redux ", and other proprietary products remain unaffected.

A. A. Jones & Shipman Limited, of Leicester, announce that they have acquired the sole selling rights for the U.K. and certain other territories of the well-known Belgian-made " Matheys " range of jig-boring machines.



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* **In the presence of dust or abrasive particles.**

Nearly 1,000,000 of these bearings are already running in a wide range of mechanisms from gas turbines to textile machinery.

Three materials with different functions have been developed to assist those who wish to design machinery that avoids the use of lubricants.

DU Thin steel strip with a porous bronze coating impregnated with a mixture of a fluoro-carbon (P.T.F.E.) and lead. Supplied in flat lengths or as finished bushes and thrust washers in a range of sizes. Available at low cost from stock.

DQ A fluoro-carbon (P.T.F.E.) strengthened with special fillers and supplied in bars and tubes. A range of diameters is available from stock. Non-standard or irregularly shaped dry bearings can be simply machined from this material.

DM A process of applying an adherent layer about .002" thick utilising the bearing properties of a combination of fluoro-carbon and molybdenum disulphide to the bearing surface of any ferrous part. Such parts must be sent to us for process.

The basic material we use in all these bearings is the "Fluon" brand of Polytetrafluoroethylene supplied to us by I.C.I. Ltd.

DU DU is a new development based on our original DP material which it now supersedes, and provides three times the load/speed carrying capacity of its predecessor.

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PERSONAL

Mr. J. C. Stevenson, B.Sc., F.R.Ae.S., has been appointed chief technician, and **Mr. E. F. Maillard, B.Sc.(Chem.), A.R.I.C., A.I.M.,** has been appointed metallurgist of Tiltman Langley Limited.

It is announced by The Consolidated Pneumatic Tool Co. Ltd. that **Mr. J. L. Ritchie, B.Sc., M.I.Mech.E.,** has been appointed personal assistant to the managing director of the company.

Mr. H. W. Boyne has joined Colloidal Graphite Limited as technical sales representative.

Mr. P. H. Jackson, M.C., has been appointed home-sales manager of the Lancashire Dynamo Group, with whom he has been for more than twenty years.

Mr. John O. Sewell has been appointed assistant general manager of the British Lock and Hardware Division and British Materials Handling Division of the Yale & Towne Manufacturing Company. Mr. Sewell will also retain his title of Comptroller.

Mr. F. Foster, M.Sc., A.M.I.E.E., has been appointed Managing Director of Crompton Parkinson (Stud Welding) Ltd., a recently formed subsidiary company of Crompton Parkinson Ltd.

Mr. A. Cheney has been appointed sales manager, and **Mr. G. F. Henderson** assistant sales manager of Achesons Colloids Limited, **Mr. T. Wint** has been appointed sales engineer of the same company, which is a subsidiary of Acheson Industries (Europe) Limited and which will shortly be moving to their new headquarters offices in Plymouth. **Mr. G. J. B. Davies** has been appointed general manager of Acheson Dispersed Pigments Company, of Dukinfield, Cheshire, and **Mr. E. A. Smith**, assistant manager, European Operations.

It is announced by the Technical Division of G. P. Wakefield Ltd. that **Mr. N. Maggs** has joined the Systems Engineering Division of the Electroflo Meters Co. Ltd. as contracts engineer. **Mr. F. Cheventre** becomes outside representative for the Systems Engineering Division in the London and Southern England areas.

Mr. M. O. L. Lynton has been appointed a director of Soag Machine Tools Ltd. **Mr. D. Nixon** becomes publicity manager.

Mr. G. A. S. HARVEY

It is with regret that we announce the death of Mr. George Alfred Sydney Harvey, President of G. A. Harvey & Co. (London) Ltd., on May 28, 1958.

Mr. H. F. BARRUS

We regret to announce the sudden death on June 4, 1958, of Mr. H. Franklin Barrus, joint managing director of E. P. Barrus Ltd.

CONTRACTS

An order worth over £1,000,000 has been placed with the Davy and United Engineering Company Limited by Northern Aluminium Company Ltd., and covers the supply of a new 4-high breaking-down and plate-finishing mill plant for their Rogerstone Works. The equipment has been ordered as part of a £10,000,000 expansion scheme to increase the rolling capacity of the Rogerstone Works from 50,000 to 75,000 tons of aluminium strip and sheet per annum.

The principal unit to be built by Davy-United will be a 38-in. and 60-in. × 144-in. 4-high rolling mill, which will be the first breaking-down mill in the Rogerstone hot-strip rolling line and will also be capable of producing finished heavy- and light-gauge plate. It will be driven by two 4000-hp motors, will have a top rolling speed of nearly 600 fpm, and will be supplied with a combined slab and plate shear of hydraulic type and all necessary auxiliary equipment. In addition, a number of Davy-United instruments will be fitted to the mill to provide means of controlling certain functions of its operation automatically and to give automatic control of the thickness of the strip rolled. Certain of the instruments will incorporate radio-active isotopes.

The Igranic Electric Company Limited, of Bedford, one of the Metal Industries Group of companies, has won a contract from New Zealand, in the face of international competition, for the supply of centralized motor control panels and auxiliary control boards for a freezing plant in Invercargill. Delivery of the equipment, which embodies several new design features and components, is scheduled for early 1959. The value of the order is about £50,000.

The British Broadcasting Corporation has placed contracts and orders valued at over £16,000 with E.M.I. Electronics Ltd., for the Company's Type-WM.2 T.V. oscilloscopes, line selectors, and ancillary units. This version of the WM.2 oscilloscope has been specially designed for monitoring television waveforms, and will be used by the B.B.C. at transmitting stations, switching centres, studios, and outside-broadcast units. Special consideration has been given to ensure that the instrument meets all essential measurement requirements associated with sine-squared pulse and bar-testing techniques, which are now being widely accepted as a standard method for rapidly checking the characteristics of video transmission lines and networks.

A £50,000 British electronic digital computer has been bought by the University of Stuttgart, in the face of strong competition from the U.S.A. The machine, a Ferranti "Pegasus", will form the basis of a new Computing Centre to be established at the University and will at first be used for research on nuclear physics, aerodynamics, and the design of precision instruments. Personnel from the University are being trained in handling the machine at the London Computer Centre.

A CROSS SPRING WASHER

is the finest locking device yet evolved!

Produced primarily for aero engines, the Cross Spring Washer is made from finest steel wire to DTD/239A specification and has the Cross patented ground edge with a perfect sprag to prevent the nut from coming loose. Even after years of vibratory conditions this spring washer will not "sit down".

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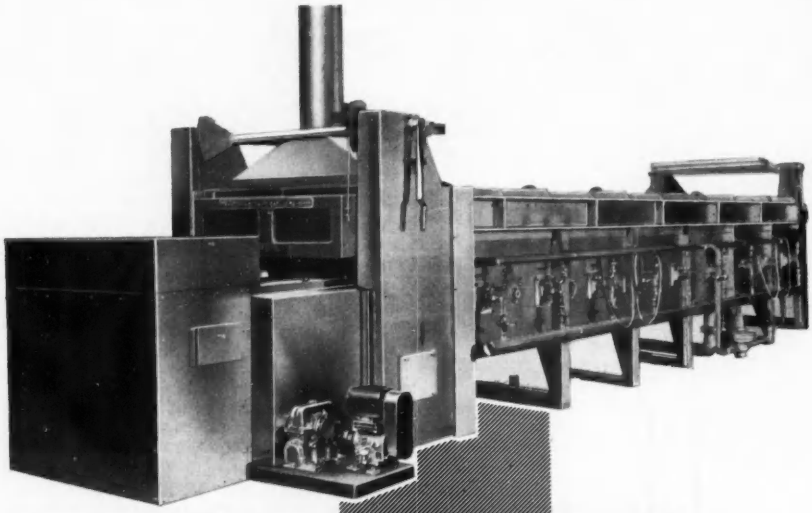


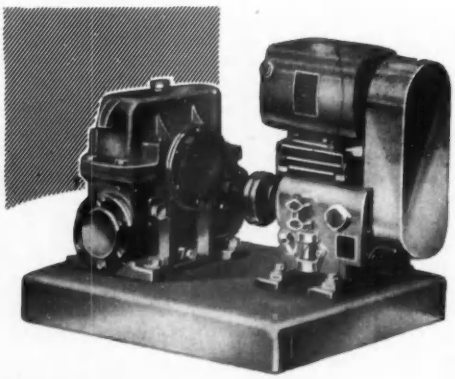
Another Carter Variable Speed Drive

This slow speed drive to a Gas Fired Continuous Tempering Furnace includes a Size F10 Carter Variable Speed Gear.

Built by the Dowson & Mason Gas Plant Co. Ltd., the process speed may be infinitely varied by means of the Carter Gear control, adjustable with the drive running or stationary.

Remote Electrical, Hydraulic, Hand or Lever controls are available with all sizes of Carter Gears.

Specialists for over 20 years in the design, manufacture and industrial application of Hydraulic Variable Speed Gears. For full details of the entire fractional up to 35 horse power range of Carter Variable Speed Drives write for Folder 658.



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The chemical-plant division of The Power-Gas Corporation Limited, Stockton-on-Tees, has obtained an order from Kali-Chemie A.G., Hanover, Germany, for the supply of a Power-Gas/Hercules reforming plant. This unit, which is similar to others erected in the U.K., is based on liquid-hydrocarbon feedstock and comprises a tubular cracking furnace followed by carbon monoxide conversion, carbon dioxide removal, and a methanation stage to provide the highest possible purity of the final hydrogen.

Gibbons Brothers Ltd., a subsidiary of Gibbons (Dudley) Ltd., have received from the South Durham Steel & Iron Co. Ltd. a contract to build the coke ovens for the new steel-making plant to be constructed as an extension of their West Hartlepool Steelworks. This contract comprises three batteries of seventy-eight of the latest type "Wilputte" ovens, with complete oven machinery, coal- and coke-handling plant, and by-product plant. In addition, orders have been received by Gibbons for the same development for four large slab-reheating furnaces and one 220-ft continuous plant normalizing furnace. The total value of these contracts amounts to more than £3,750,000.

Mirrlees, Bickerton & Day Limited, of Stockport, a member of the Hawker Siddeley Group, have supplied an auxiliary generating set for the Georgia Generating Station at present being constructed for the British Columbia Power Commission at Chemainus, B.C. The set consists of a Mirrlees JVSS16 diesel engine, continuously rated at 2105 bhp at 900 rpm, with a 10% overload rating of 2315 bhp for one hour, driving a Brush 1500-kW alternator. The engine is arranged for auto-starting and shut-down and is complete with all the necessary automatic features, together with audible and visible alarm gear. The diesel generating set will be used to provide the current required by the electric motors for starting the main thermal generator units.

TECHNICAL ASSOCIATION OF INVESTMENT FOUNDERS

A new technical association, serving the investment-foundry industry using expendable-pattern techniques for the production of industrial metal castings, has been formed under the title of The British Investment Casters' Technical Association.

The objects and scope of the new organization are entirely devoted to technical aspects of investment casting and will include such activities as the preparation of specifications for materials and testing procedures, the improvement of production techniques, expansion of the application of investment castings, and the general exchange of technical information within the industry.

NEW FOUNDRY FOR SAMUEL OSBORN & CO. LIMITED

In planning their new Holbrook foundry, Samuel Osborn & Co. Limited, of Sheffield, decided not only to cater for future expansion but also to cater for the latest provisions of the Factory Act and Foundry Regulations. As a result, virtually ideal working conditions have been achieved and provide a degree of cleanliness which seems quite remarkable in what is usually considered a dirty industry.

The foundry, which caters for the production of the "Socast" steel castings made by the precision techniques developed by the company, and comprising the shell-mould, Osborn-Shaw, and Osborn CO_2 block processes, consists of two bays of 50 × 200 ft, and a service annexe of 30 × 200 ft. There is a consecutive production flow, starting with Shaw moulding, shell moulding, CO_2 moulding, core-making for shells, shell closing, casting, melting, cut-off, initial shot-blast, stand-grinding, finish-fettling, heat treatment, shot-blast, inspection, and dispatch. Provision has been made for the temporary accommodation of the CO_2 and Osborn-Shaw processes, but the factory has been ultimately designed for the shell-mould process only.

The fettling shop has been designed to have adequate mechanical handling facilities, using roller track and bins. Dust extraction is achieved by two fans, each of which is capable of extracting 30,000 cfm. In this connection, a recent dust count in the fettling shop showed that there was less dust present than in the heart of the Sheffield industrial area.

The offices have been laid out to provide adequate locker- and wash-rooms, and the canteen is at present housed in a space which will eventually be allocated for these rooms. There is also provision for a second storey to the building, when justified by future expansion.

ANALYSIS OF THE EUROPEAN ELECTRICITY-SUPPLY INDUSTRY

Every two years, the Electricity Committee of the Organization for European Economic Co-operation analyses the situation of the European electricity-supply industry in a general report. These reports provide an opportunity for a thorough discussion of the main problems affecting the industry. In 1955, for instance, the economic factors influencing the development of nuclear energy were discussed. The present report lays special stress on the long-term outlook up to 1975 and is divided into five parts. Of these, parts I and II deal with the present situation of the industry; part III discusses the medium-term prospects and the difficulties confronting the development of the industry, and part IV touches on the long-term prospects. The last pages are devoted to a review of the Committee's principal activities in 1956 and the part it played with regard to international co-operation.

The statistical data and the medium- and long-term estimates used for compiling this report were supplied by the countries concerned. All this information is reproduced in an appendix containing seven series of tables on plant capacity, maximum available capacity, output of public utilities and auto-producers, distribution of consumption by sectors, and fuel consumption in power stations. Until this year it had not been possible to give an analysis of the production and consumption of auto-producers, but, thanks to the work achieved by the International Federation of Industrial Auto-producers and Consumers of Electricity (FIPACE), which now publishes an annual Statistical Bulletin, detailed figures have been obtained for most countries.

The report shows that the electricity-supply industry has continued to expand at a yearly rate of about 8% in 1956, and has reached twice the level or production of 1948. No increase on this scale had even been recorded over such a period in the past. To meet an ever-increasing demand, the electricity undertakings in the O.E.E.C. area have launched investment programmes based on an average increase in consumption of 7% per year for the next five years. Most of them, however, fear that they will be unable to carry out these programmes entirely, owing to the difficulties encountered by electricity producers in raising the necessary funds. Rising interest rates on the money market, combined with the often very stringent price controls imposed by governments, are the two difficulties most frequently mentioned. These circumstances make increases in electricity rates likely. Any increase should, however, be slight, owing to constant improvements which reduce the cost of equipment in terms of real value and which save fuel.

The member countries have also given some guidance on the long-term trend of electricity requirements and on how these requirements may be met. According to their forecasts, consumption might be roughly three and a half times larger in twenty years' time. This period will witness a gradual tailing off in the development of hydro-resources, owing to the exhaustion of the more profitable sites, whilst nuclear energy will begin to develop on a large scale. The rate of expansion of nuclear production

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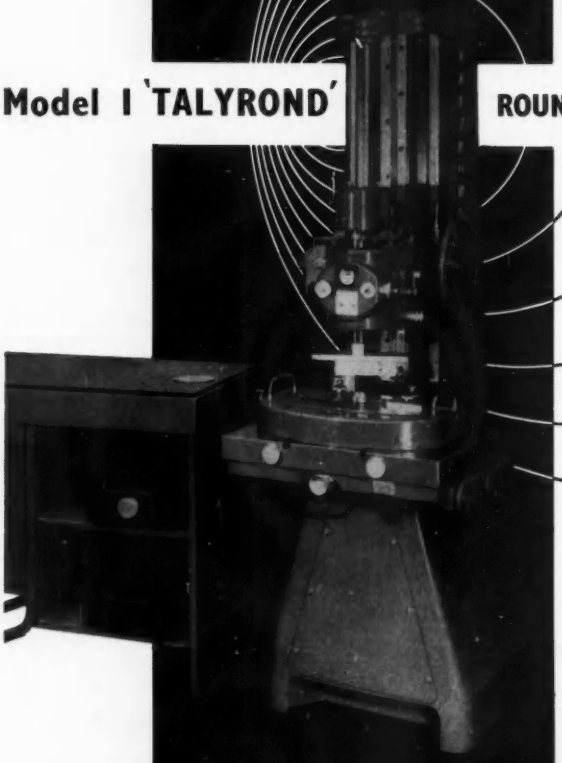

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will depend on technical and economic factors which are still uncertain ; the first estimates made by the countries show that 30 to 35% of the total thermal energy produced in Europe might be of nuclear origin by 1975, and this would mean a fuel saving of about 120-million tons in terms of coal equivalent. The achievement of these aims will also depend on whether the capital required can be raised on reasonable terms.

THE BRITAIN-IN-EUROPE COMMITTEE

The Britain-in-Europe Committee is a new organization formed by leading figures from all sides of industry, finance, and commerce, to stimulate interest in Britain's participation in the proposed European Free-Trade Area. It owes its inception to the interest aroused by the *Economist* Intelligence Unit's study "Britain and Europe", commissioned by the Britain and Europe Fund. All but six of the hundred subscribers to the Fund agreed that the surplus should be vested in The Britain-in-Europe Committee.

The main function of the Committee will be to undertake or sponsor original research on European economic integration, especially with regard to the proposed European Free-Trade Area, and to make known the results. The Committee, which is incorporated as a company limited by guarantee, will also provide specialized information and documentation on European economic integration, besides initiating or supporting courses, lectures, and meetings.

The offices of the Committee are at 61 Catherine Place, Palace Street, London, S.W.1 (Tel.: VICtoria 4165/6).

BRITISH STANDARDS

(*Copies of British Standards may be obtained from the British Standards Institution, 2 Park Street, London, W1*)

Wrought-Iron and Mild-Steel Hooks (B.S. 482 : 1958).

In this revised Standard (price 8*s.* 6*d.*), six types of hook are dealt with, i.e., trapezoidal and circular-section shank hooks, trapezoidal shank hooks with integral collars, trapezoidal circular-section eye hooks for use with chain or with shackle pins (B.S. 825), and trapezoidal eye hooks for use with wire-rope thimbles (B.S. 464).

Material, form and dimensions, screw threads, heat treatment, proof testing, inspection, and marking are specified. The safe working loads of the largest hooks are 50 tons for trapezoidal shank hooks, 13½ tons for trapezoidal eye hooks, and 5 tons for circular hooks.

Appendices give the information which should be supplied with the inquiry and order, together with a certificate of test and examination, details of Whitworth and Unified screw threads, and notes on design. As compared with earlier editions, the recommended finished machine diameters for shanks have been added and the diameter at the top of the thread has been designed to give an average stress of not more than 3 tsi at the minimum core diameter of the thread.

Dimensions of Poppet Valves, Collets, Valve Guides, Valve-Seat Inserts, and Tappets for I.C. Engines (B.S. 2988 : 1958).

Intended to provide a basis for new designs, this British Standard (price 6*s.*) has been prepared from a summary of dimensions used in existing practice. It provides for some measure of rationalization and interchangeability between poppet valves, ordinary and heavy-duty poppet-valve-seat inserts, poppet-valve guides, poppet-valve collet assemblies, and mushroom, barrel, and cylindrical types of tappets. This is achieved by specifying ranges of principal valve component dimensions whose increments vary throughout their ranges in the manner considered to be most appropriate. Throat diameters of valves range from ⅞ to 6⅞ in., and valve lengths from 3 to 17 in.

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NEW TECHNICAL BOOKS

The Practical Engineer Pocket Book. 66th Edition, 1958. Edited by N. P. W. Moore, B.Sc., A.C.G.I., D.I.C., A.M.Inst.F., A.M.Inst.P. 788 pages. Publishers : Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2. Price : 12/6.

Since its initial publication in 1889, this handy little reference book has enjoyed wide popularity, combining as it does a wealth of useful information, concisely expressed and conveniently presented, in easily readable form. This new edition is no exception and worthily follows the example of its predecessors.

Since the publication of the previous edition in 1954, various technical advances have been made in many fields. As a result, this 66th edition incorporates a number of modifications to various sections, while the section on metallurgy has been largely rewritten, and the properties of creep and fatigue have been briefly introduced. In addition, the section dealing with the thermodynamic properties of steam has also been extensively recast, and new tables of steam properties devised. As always, this profusely illustrated pocket book includes a comprehensive index, a detailed list of British and American journals, a useful technical dictionary of German-English, French-English, and Spanish-English terms, and numerous conversion and other tables.

Practical Engineering Metrology. By K. W. B. Sharp, B.Sc., A.M.I.Mech.E., A.M.I.P.E., F.R.S.A. 256 pp., 164 illustrations. Publishers : Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2. Price : 30/–.

Chiefly intended for students of metrology, and fulfilling the need for a book in which basic principles are given adequate treatment, this well-written volume mainly covers fundamentals on which students can build, describing a variety of methods which will be of use in future applications, and pointing out the importance of knowing the accuracy to which measurements can be made by the various methods and instruments. In addition, it covers a considerable proportion of existing examination requirements.

Quite apart from its undeniable value to students, this excellent book should also be of definite service to inspectors in industry and to teachers, by providing ideas for practical work and by including certain useful information not previously published.

Oral Communication of Technical Information. By Robert S. Casey. 204 pp., 20 illustrations. Publishers : Reinhold Book Division, New York, and Chapman & Hall, Limited, 34 Essex Street, London, W.C.2. Price : 36/–.

This beautifully produced volume is undoubtedly one of the most interesting and useful handbooks of its kind ever published, and, in addition, has the merit of making fascinating reading. Covering methods of effective speaking by technical men to other technical men and also to laymen, for most situations, including demonstrations, lectures, and legal testimony, it indicates the bridges and barriers to effective technical communication, and then takes up the actual problems involved and how to solve them.

In addition to valuable and pertinent information on how to preside at meetings, and how to discuss scientific matters socially, in the lecture room, or in court, the book covers the organization of material, composition, and the delivery of formal and impromptu speeches. In addition, much useful information is included on the use of mechanical aids, including sound recordings, teleprompter equipment, and slides, with useful recommendations regarding their content and presentation, together with examples illustrating how to avoid insufficiency or " overcrowding " of information in visual aids.

Analytical Mechanics for Engineers. Fifth edition, 1958. By Fred B. Seely, Newton E. Ensign, and Paul G. Jones. 490 pp., 569 illustrations. Publishers : John Wiley & Sons, Inc., New York, and Chapman & Hall Limited, 37 Essex Street, London, W.C.2. Price : 58/–.

Presenting those principles of mechanics which are essential for the study of engineering, Professors Seely, Ensign, and Jones build up these principles as far as possible from common experience and apply them to the solution of problems frequently encountered in engineering practice.

As in previous editions, this new edition is divided into four parts, i.e., statics, kinematics, kinetics, and special topics, the latter section including such subjects as flexible cables, vector analysis, the motion of a point relative to a rotating set of axes, and mechanical vibrations. Also, as before, emphasis is placed on the use of the general steps or procedure in analysing problems in equilibrium and dynamics. In addition to a new chapter introducing vector analysis, this edition includes a chapter on the study of equilibrium of bodies, making use of the concepts of virtual work, and a separate chapter on the use of the inertia-force method in kinetics, while a general coplanar force system is used in introducing equilibrium of force systems. Other material in the new edition includes the subject of the product of inertia of areas and the addition of many new problems involving actual engineering or physical conditions.

Pre-Stressed Concrete, Theory and Design. By R. H. Evans, C.B.E., D.Sc., D.ès Sc., M.I.C.E., M.I.Struct.E., and E. W. Bennett, M.Sc., A.M.I.C.E. 310 pp., 110 illustrations. Publishers : Chapman & Hall Limited, 37 Essex Street, London, W.C.2. Price : 60/–.

Based on the belief that principles are more important than details, and therefore omitting much of the purely descriptive material usually found in books on prestressed concrete, this authoritative and valuable book caters for both students and practising designers, as well as engineers.

Part I of the book discusses the basic principles applying to all types of prestressed-concrete structures. Wherever possible, principles are established theoretically, but constant reference is made to experimental work, and examples are provided for students to work out. Part II deals with the design of simply supported beams, which constitute the greater part of the prestressed work hitherto completed. Here, the aim is to bridge the gap between principles and practice, using the former as the basis of a procedure that can be carried out in the design office. The maximum possible use is made of formulae, tables, and graphs. Part III provides an introduction to some of the more specialized types of prestressed structure.

Conductance-Curve Design Manual. By Keate A. Pullen, Jr., Eng.D. 128 pp., 78 illustrations. Publishers : John F. Rider Publisher, Inc., 116 West 14th Street, New York 11. Price : $4.25.

Utilizing an original technique for designing electronic circuits, thereby shortening design time and minimizing problems, this useful manual provides for the benefit of engineers, scientists, and technicians considerable data organized to help users design circuits which will function with a minimum of readjustment. The manual is divided into three main sections, i.e. (1) a brief explanation of the special curves incorporated and their application in typical R.C. amplifier design, (2) a set of tables for use in making tube substitutions, and tables for simplifying the selection of tubes for given applications, and (3) a special set of curves for facilitating tube-circuit design.

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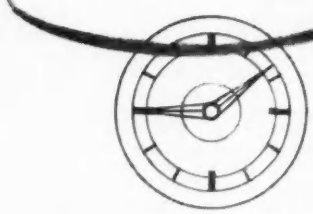

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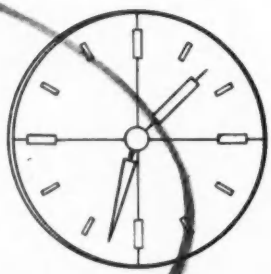

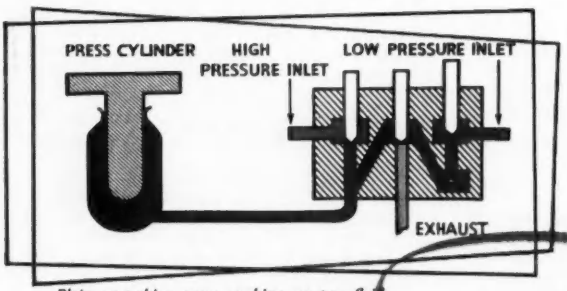

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2. Extrusion Press Components. Produced by a firm which has specialized in this work for many years and which has the skilled judgment necessary for the determination of suitable steels and hardness levels, a neatly compiled illustrated 24-page brochure outlines the problems encountered with components in the construction and operation of extrusion presses.

The components dealt with include containers, liner holders, liners, rams, dies, die holders, bolsters, mandrels, mandrel holders, pressure pads, scraper plates, and crosshead castings, and a section is included with tabulated data on steels recommended for the manufacture of extrusion components.

3. Wire Drive for Milling Cutters. A 4-page leaflet briefly describes an improved wire drive for milling cutters, etc., consisting of a shaft or arbor made from a high-tensile case-hardening steel with a tangential cut for the round wire spiral, the latter being drawn from spring steel. Although breakages are very rare, the wire spiral is replaceable. An advantage of the tangential cut for the wire is that shearing is eliminated, and the wire is in compression.

The loose details so necessary with some conventional attachments are not used with this drive. On the contrary, the cutter is swiftly attached or replaced ; there is accuracy of bearing on the walls and of the shoulder of the arbor against the cutter. Furthermore, the full face of the cutter is used, and cutters and arbors are interchangeable.

4. Electro-Lifting Magnets. In a well-illustrated 12-page brochure, descriptions and specifications are given of circular and bi-pole electro-lifting magnets, together with illustrations of several mobile cranes fitted with these magnets.

The circular magnets described are suitable for general use, as their deep-penetrating magnetic field enables them to handle all kinds of scrap and other forms of metal where careful stacking and arrangement is not of prime importance. The bi-pole magnets are especially suitable for lifting billets, ingots, rails, and similar metal sections, will handle hot or cold materials with equal ease, and will lift single billets or rails out of a layer without upsetting others, or a number of billets without disturbing the underlayers. Details are also included of magnet control gear and cable coiling drums.

5. Portable Welding Generator Unit. Details of a portable welding generator unit powered by and mounted on the " Land Rover " are given in a 2-page leaflet. This unit has been specially designed as a portable package-type 150-A d.c. welding unit and is driven directly from the vehicle by a simple, efficient belt drive. The generator is a single-operator, variable-voltage type with laminated pole-pieces, and output can be varied both as to voltage and current independently.

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6. Aluminium Alloy for High-Speed Machining. Developed to meet the problem of disposal of swarf when machining aluminium alloys on high-speed automatic lathes, a special heat-treatable aluminium alloy in the medium-strength range is described in a 4-page leaflet. This alloy is available in the form of extruded and heat-treated machining bar and also as extruded forging bar.

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7. Moisture Meter. A 12-page brochure fully describes a moisture meter and discusses properties of the material which it may be required to test, so that it is possible to judge whether or not the meter is likely to solve a particular moisture problem.

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8. Valves, Cocks, and Pipeline Fittings. A comprehensive catalogue arranged in a form which facilitates quick reference to any item gives detailed information on a range of valves, cocks, and pipeline fittings, including gate valves, globe valves, oblique valves, check valves, line blind valves, oil valves, relief valves, level gauges, hygienic fittings, and fire-protection equipment. Illustrations of the various items are included, together with reference tables of standard pipe threads and flanges and of corrosion-resistance.

9. Air-Cooled Heat Exchangers. Designed for small and large industrial cooling duties, the heat exchangers described in a 14-page well-illustrated brochure normally operate by passing air across finned tubes which carry the fluid to be cooled, thus dissipating waste heat to the atmosphere.

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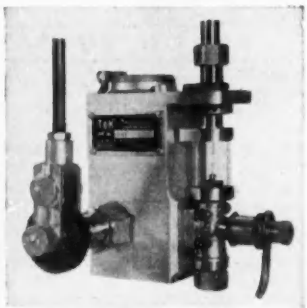

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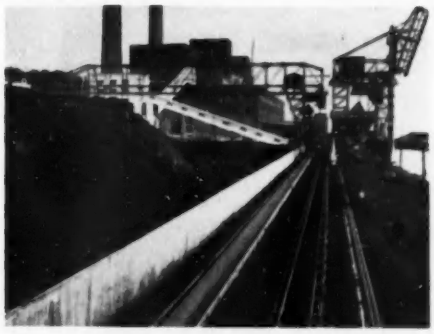

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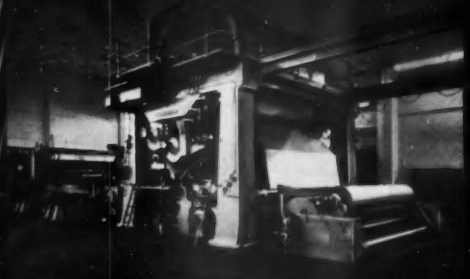

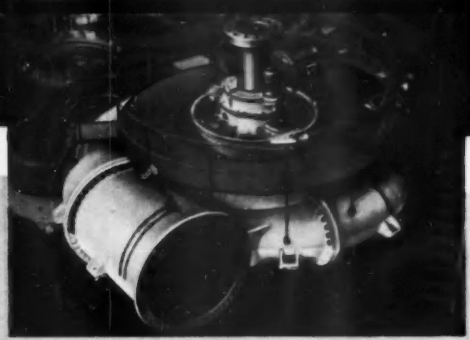

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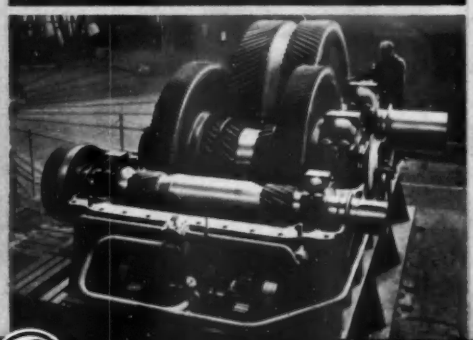

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Brockhouse, J., & Co. Ltd. A109
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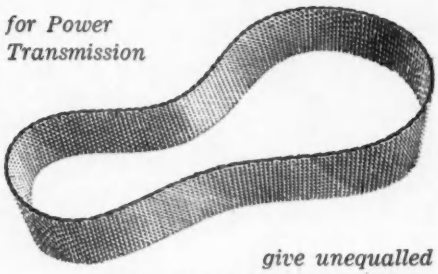

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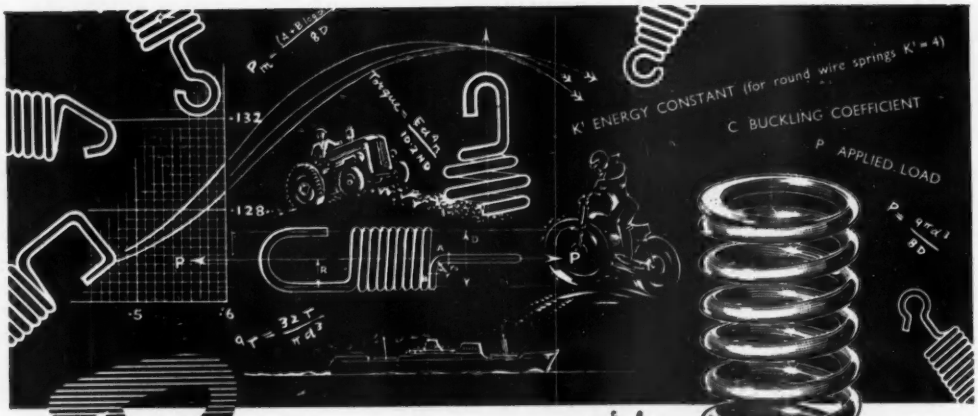

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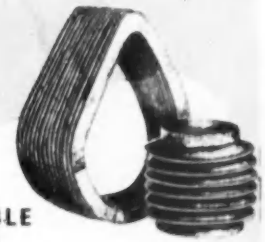

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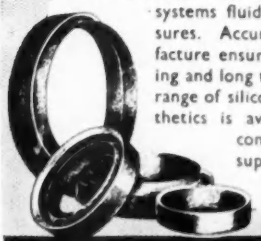

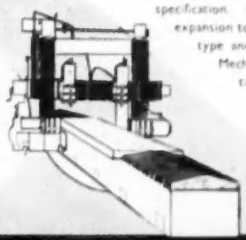

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